Refrigerating Machines:

Compression,
Absorption.

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Compression, Absorption.

Comparison of Capacities and Economies of Compression and Absorption Systems, and of Combined Compression and Absorption Systems.

Being the Complete Text, Figures and Diagrams of the Paper entitled "Comparison of the Production of Cold by Compression of Liquefiable Gases and other Methods," read before the FIRST INTERNATIONAL CONGRESS OF THE REFRIGERATING INDUSTRIES, PARIS, 1908, and published with the consent of Sec. General J. de Loverdo, F.I.C.R.I.

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PREFACE.

AT the request of the American Committee for the First International Congress of Refrigerating Industries, the author consented to prepare a paper for the Congress on the subject—"The Production of Cold by the Compression of Liquefiable Gases and other Methods."

During the author's continuous practice of refrigerating engineering, since 1891, he has accumulated an extensive mass of data from his practical experience and from numerous tests of compression and absorption systems of refrigeration.

Although such data is usually reserved as a private personal asset, the author was tempted to and has given in this paper a great deal of such data that he would not have done for an occasion less momentous than an International Refrigerating Congress. The data as given should enable refrigerating engineers to more readily, and with less labour on their parts, solve all problems of capacity and economy of a compression or an absorption system, or of their combinations. Through the kindness and courtesy of Sec. J. De Loverdo of the Congress, the author has his permission to publish this paper in advance of the congressional publications that will appear at a later date. The author hopes that his humble efforts may prove of value to his brother refrigerating engineers, and that they may ever strive to give out their own personal data for the general good of the Refrigerating Engineering Fraternity.

GARDNER T. VOORHEES.

Refrigerating Machines:

Compression, Absorption.

THIS subject is, in effect, the mainspring of mechanical refrigeration, as without the machine to produce the refrigeration, there is no necessity of discussing how, when, or where to apply such refrigeration. The so-called production of cold is indirectly accomplished by the compression of liquefiable gases, in that the gases, through the process of compression, are prepared for condensation, and are as liquids re-evaporated or expanded into their gases, thereby producing refrigeration, which is commonly called cold.

First, let us take up the production of refrigeration (cold) by the evaporation of volatile liquids and the reliquefication of their gases by compression.

Cold is a relative term used only in comparing the temperatures of two bodies; one body may be cold when compared with another body, or hot when compared with still another body. Boiling water on a hot stove is cold compared with the stove, and hot compared with ice. Water will be cooled by a block of ice, and anhydrous ammonia heated by the block of ice. Practically all known substances have three distinct states, which are solid, liquid, and gaseous. In each state the substance is either hot or cold as compared with its temperature in another state.

Everyone is familiar with the three states of water, which are ice, water, and steam, the solid state as ice being the coldest, the liquid state as water being the next hottest, and the gaseous state as steam being the hottest.

Likewise many other substances have their three states as solid, liquid, and gaseous, with like relative degrees of heat or cold, from where the liquid is very hot to where it is very cold, as, for example, from platinum to hydrogen. Such a scale of substances and the temperatures of the freezing and boiling points is as follows: Each substance is a vapour or gas after being heated above its boiling point, and a solid after being cooled below its freezing point. All temperatures in this paper are hereafter given in Fahrenheit degrees.

Substance.	Freezing Point.	Boiling Point.
Cast-iron ...	2200°	3300°
Water	32°	212°
Air	—340°	—312°

Here we have solid water, solid iron, and solid air; liquid water, liquid iron, and liquid air; and gaseous iron, gaseous water, and gaseous air. We might speak of them as iron ice, water ice, and air ice; or water iron, water water, and water air; or iron steam, water steam, and air steam. Using such expressions, a block of water ice is cold to a block of iron ice, and hot to a block of air ice.

Hereafter when the word vapour is used it generally means the gaseous state of a substance at the temperature of its boiling point, and when the word gas is used it generally means a vapour that is superheated or

above the temperature of its liquid's boiling point, all at the pressure of the liquid's boiling point.

In this paper we are not directly concerned with the solid states of substances, as that properly comes under the head of ice making. Briefly stated, we have here to deal with refrigerating machines, and this paper will confine itself to machines that produce refrigeration through the expenditure of heat as such or through the expenditure of mechanical energy, or through a combination of both. In all such classes of refrigerating machines, the refrigeration is produced either by the evaporation or expansion of a volatile liquid or by the expansion of a gas while doing external mechanical work. Such machines can be divided into two great classes:

Vapour Machines.
Gas Machines.

For brevity, let us omit all historical mention of the various machines, as such information is not of material moment to this paper, and is available in the many books on this subject.

Vapour machines are those wherein refrigeration is done by the evaporation of a liquid into its vapour. In some of these machines the vapour so formed is thrown away, but in most cases the vapour is reliquefied.

In the gas machine the refrigeration is produced by compressing a gas, then abstracting its heat of compression, and then expanding it behind a moving piston that does external mechanical work; the gas in doing external mechanical work loses part of its heat energy, which is transformed into mechanical energy, so that the gas is cooled by its loss of heat energy. Machines

of this class are usually operated with air as the refrigerating medium, and, although formerly used extensively, are now, in the United States at least, but little used because of their supposed inefficiency as compared with vapour machines.

Gas machines, commonly called cold air machines, usually compress the air and cool it after compression to within 20° of the temperature of the cooling water available, and then expand the compressed air in an engine. The expanded cold air is then circulated either directly through chambers to be cooled or through a closed pipe circuit directly to the suction inlet of the air compressor cylinder. In the cold air machine with the closed pipe circuit much of the inefficiency and loss due to moisture in the air is eliminated. The cold air from machines of this class often leaves the engine cylinder at from 50° to 80° below zero.

VAPOUR MACHINES.

Were it not for the cost of the volatile liquid all such machines would be extremely simple, as they would only require that a volatile liquid should be evaporated in a refrigerator and the resultant vapour thrown away. It is the recovery of this vapour that calls for the so-called refrigerating machine.

Vapour machines can be divided into two classes—Mechanical Compression Machines, hereafter called Compression Machines, and Heat Compression Machines, hereafter called Absorption Machines. All of these machines depend upon the physical properties of a volatile liquid in taking up the latent heat of vaporisation when it changes its form from a liquid to a vapour. Everyone is familiar with the action of water, which

would make a most ideal volatile liquid for refrigeration if its temperature place in the scale of substances were not too high for general practical use. Water at atmospheric pressure requires approximately 965 British Thermal Units (hereafter called B.T.U.) to change its state from liquid to vapour. If it were desired to refrigerate any substance at a temperature of, say, 300°, then water at atmospheric pressure would evaporate at 212° and refrigerate said substance. In many cases, however, all of the refrigeration of a plant is indirectly done by the evaporation of water, as when the cooling water for the condensers is cooled by its evaporation in a cooling tower. As we see, water at ordinary pressure is not what we look on as a refrigerant liquid, so other liquids having lower boiling points at atmospheric pressure are commonly used, such as sulphuric ether with a boiling point of 96°, sulphur dioxide with a boiling point of 14°, anhydrous ammonia with a boiling point of —29°, and carbonic acid with a boiling point of —124°, etc. Each of these refrigerant liquids has its advantages and disadvantages over the other refrigerant liquids mentioned. Some of them attack one metal, others do not; some require excessively high condenser pressures, others excessively low refrigerator pressures; some large compressor cylinders, some small compressor cylinders; some of the vapours are comparatively harmless when breathed by man, while the vapours of others when breathed even in small quantities are often fatal; some are inflammable, and when mixed with air are explosive, others are not; and so on.

The purposes of this paper will be best served in not going further in detail into the desirability or undesir-

ability of one refrigerant fluid over another, except to say that ammonia and carbonic acid are the refrigerants most used to-day, ammonia being greatly the predominant refrigerant in use.

It is well known that the boiling point of a liquid, which is the same as the condensing point of its vapour, is raised or lowered as its pressure is raised or lowered; herein lies the principle of vapour machines.

The liquid is evaporated at a low pressure, which does refrigeration by taking up latent heat in evaporating at a low temperature; the vapour so formed is compressed to a high pressure, so that its condensing point is above the temperature of the available condensing water.

Heat, like water or electricity, only flows from a higher level to a lower level, and the sole object of the refrigerating machine is to elevate the heat abstracted at a low level to a level sufficiently high so that it can be dumped into a convenient waste heat drain.

Just as one would remove water from a flooded cellar by lifting it with a pump and discharging it into the street gutter, so the vapour refrigerating machine lifts the heat absorbed at the low temperature to a higher temperature, where it can be dumped into the condensing water, so that in reality the refrigeration done is indirectly accomplished by having a supply of condensing water to dump heat into.

The heat taken from a pound of water at, say, 32° in freezing it into ice is dumped into condensing water at, say, 70°, which raises the temperature of, say, 14.4 pounds of water to 80°; this is without considering the heat of compression which will be taken up later.

It seems to be well established that so long as a refrigerant fluid is adapted to the temperature required, it is not material what refrigerant liquid is used, so far as the efficiency of the machine is concerned, when the refrigerant fluids work between fixed refrigerator and condenser temperatures.

The Vacuum Machine may be either of the compression or absorption type or a combination of both. In its simplest form it does refrigeration by the evaporation of water at a pressure very near a perfect vacuum, but as the water vapour so formed is of great volume, it requires very large compressor cylinders to exhaust it, and to discharge it into the atmosphere, so an absorbent for water vapour as sulphuric acid is used, and so the compressor cylinder is reduced to one of comparatively small size, having the function to remove the air that is left after the water vapour has been absorbed by the sulphuric acid. Machines of this type are but very little used in the United States.

A machine of this type, called the Patten machine, is used in a few instances, and ice-making plants of this system of some considerable capacity have been erected, and their successful operation has been witnessed by the author.

In absorption machines the operation is parallel to that of the compression machine, except that the ammonia vapour from the refrigerator is absorbed in an absorber by weak aqua ammonia, thereby giving strong aqua ammonia. The strong aqua is pumped from the absorber into a still, wherein it is heated, and the ammonia vapour is distilled from the strong liquor into a condenser, and the weak liquor is returned to the absorber. *The absorber is equal in effect to the*

suction stroke of the compressor, and the still is equal in effect to the compression stroke of the compressor.

This paper will deal only with ammonia as the refrigerant fluid. Anhydrous ammonia has a varying latent heat of evaporation and condensation due to its pressure. In this paper the ammonia tables of Wood as recalculated by Davidson have been used, as given in the "Compend of Mechanical Refrigeration," by Siebel, published by the Nickerson and Collins Co., of Chicago, U.S.A. The specific heat of liquid ammonia is assumed to be as one, and the specific heat of ammonia gas at constant pressure is assumed to be .532, both as compared with the specific heat of water as one.

The following is a part of the report of the committee of the Am. Soc. Mech. Engs. for suggesting a standard tonnage basis for refrigeration: "The unit adopted to measure the cooling effect, or the refrigeration, is the heat required to melt one pound of ice, which is 144 British thermal units, and by dividing the refrigeration measured in B.T.U. by 144 the ice-melting capacity in pounds is obtained. The unit for a ton (2,000 lbs.) of ice-melting capacity is therefore 288,000 B.T.U. . . . The commercial tonnage capacity is the refrigerating effect expressed in tons of ice-melting capacity produced by a machine in twenty-four hours when running continuously under the standard set of conditions. . . . The best set of conditions to adopt seems to be those which often exist in ice-making, namely, that the temperature of the saturated vapour at the point of liquefaction in the condenser be 90° Fahrenheit, and the temperature of

evaporation of the liquid in the refrigerator be zero degrees Fahrenheit."

The committee has gone further in detail at some length in this matter, and has submitted other preliminary reports, and has proposed rules for test, etc., but as yet the final report on the subject has not been rendered.

From the above we have 288,000 B.T.U. per twenty-four hours as a ton of refrigeration. Zero degrees Fahrenheit equals refrigerator temperature, equals 15 lbs. per square inch gauge approximately. 90° Fahr. equals condenser temperature, equals 170 lbs. per square inch gauge condenser pressure approximately. As there are 1,440 minutes in 24 hours, a ton of refrigeration = $\frac{288000}{1440}$ = 200 B.T.U. per minute to equal one ton of refrigeration per 24 hours.

In this paper these figures will be used for a standard ton of refrigeration for comparison—that is, we will use 200 B.T.U. per minute per ton of refrigeration per 24 hours, and assume a standard ton of refrigeration to be done at 15 lbs. refrigerator and 170 lbs. condenser gauge pressures in pounds per square inch above the atmosphere.

From the ammonia tables the latent heat of evaporation and condensation at various pressures and the volume of a cubic foot of vapour at various pressures and the relation of the boiling point and pressure can be found, and have been used as from these tables (Compend Tables before mentioned) in all the work that follows.

Ammonia in doing refrigeration first flows as a liquid from the condenser through an expansion valve, where its pressure is reduced from that of the condenser to

that of the refrigerator, in the standard condition from 170 lbs. to 15 lbs. The temperature of the liquid to the expansion valve is 90°, and after the expansion valve at 15 lbs. is zero degrees. This reduction in temperature of the liquid ammonia is caused by the evaporation of a portion of the liquid, which thereby through its latent heat cools the balance of the liquid to the refrigerator temperature of zero degrees. From the tables the latent heat of evaporation at 15 lbs. is 556 lbs. B.T.U. per pound, to cool the ammonia from 90° to 0° or through a range of 90° requires 90° — 0° = 90°, and 90° × the specific heat of liquid ammonia or 90° × 1 = 90 B.T.U., so that the nett B.T.U. left available to evaporate the balance of the pound of liquid ammonia to its vapour to do refrigeration is 556 — 90 = 466 B.T.U., and as 200 B.T.U. per minute are required to produce one ton of refrigeration, the pounds of ammonia per ton of refrigeration per minute are $\frac{200}{446}$ = .43 lb. per minute. The amount of liquid evaporated in cooling the liquid from 90° to 0° is $\frac{90}{556}$ = .162, so that as the ammonia enters the refrigerator it is practically 16% vapour and 84% liquid. The pounds of ammonia per minute per ton of refrigeration and the percentage of liquid evaporated at the expansion valve are given in the following table, which results were obtained as above described. All pressures are gauge pressures and all temperatures Fahrenheit degrees.

The method of applying the cooling effect of ammonia varies greatly, but this general principle can be laid down as giving the best results for the least square feet of surface of pipe used in the refrigerator,

have as much ammonia liquid and as little ammonia gas in contact with the cooling coils as is possible.

The reason for this is that gas transmits heat only about 1-30th as fast as does liquid.

Also *have the substance to be cooled (as say brine) in rapid circulation in contact with the surface cooled by the ammonia liquid.*

Condenser pressure Condenser temperature		140 lbs. 80°	170 lbs. 90°	200 b 100°
Refrigerator pressure Refrigerator temperature	0 lbs. —29°	.431 lb. 19.0%	.441 lb. 20.8%	.451 lb. 22.5%
Refrigerator pressure Refrigerator temperature	15 lbs. 0°	.420 lb. 14.4%	.430 lb. 16.2%	.440 lb. 18.0%
Refrigerator pressure Refrigerator temperature	30 lbs. 17°	.415 lb. 11.6%	.425 lb. 13.4%	.434 lb. 15.2%

The best refrigerator of to-day is the brine cooler, where the ammonia liquid boils in contact with pipes through which brine to be cooled is rapidly circulated. There are many types of such brine coolers, and all have certain advantages or disadvantages one over the other, which will not be discussed in this paper. The next nearest approach to the brine cooler in efficiency of pipe surface is the flooded system of Krebbs, where the ammonia liquid is as free from vapour in the pipe coils as is possible. The ordinary expansion coils are usually governed by an expansion valve to each coil, which require much attention to regulate, and which give inefficient results as compared with the same surface in a brine cooler or in a flooded system.

The general way to operate such expansion coils is, with downward expansion for absorption machines, to

take care of the water that is entrained in many absorption machines, and upward or downward expansion for compression machines. Ammonia liquid and gas flow down the coil with downward expansion and up the coil with upward expansion. In the flooded system the coil is usually fed from the bottom, and is as full of liquid as it can be, allowing for the gas that must necessarily be there, that is formed during the evaporation of the liquid in the coil.

The author's experience is that, owing to the many practical difficulties in operating direct expansion plants, *more capacity and better economy* are realised whether the plant be for refrigeration or ice making by operation *with a brine cooler than by any other way now known to him.*

What has been said above applies either to the compression or the absorption system, and we are now supposed to have done refrigeration by evaporating liquid ammonia in the refrigerator into its vapour, and now wish to reliquefy the vapour so as to re-use it in the form of a liquid. This will be taken up under two systems, the compression system and its modification, the Multiple Effect Compressor System, and the absorption system, and a combination of the compression and absorption systems and a combination of the multiple effect compressor system and the absorption system.

In the compression system the vapour from the refrigerator is pumped into the condenser by a gas pump.

There are many forms of such gas pumps called compressors, such as single acting vertical, double acting vertical, double acting horizontal, wet or dry compression. The same general principles apply to all,

which are that during the suction stroke of the piston the ammonia vapour is drawn into the cylinder past a suction valve, and during the compression stroke the ammonia gas is compressed and discharged past a discharge valve into the condenser. A prime requirement for all types of compressors is that there shall be little or no clearance at the end of the compression stroke in the cylinder, for if there is the gas so held in the clearance space will re-expand like a piece of compressed rubber on the next suction stroke, and so prevent a full cylinder of new vapour from being taken in on the next suction stroke.

The areas of the suction and discharge pipes and the suction and discharge valves should be ample, so as not to have a materially lower suction pressure or higher discharge pressure in the cylinder than exists in the refrigerator or the condenser, for a reduced suction pressure in the cylinder means reduced capacity, and an increased discharge pressure means increased power to drive the compressor. The piston and valves should be tight as against leakage of ammonia passed or through them.

The Dry Compressor takes the vapour or gas from the refrigerator and compresses it and discharges it into the condenser, and the compressor cylinder is surrounded by a water jacket to take up part of the heat of compression. The Wet Compressor takes in the vapour and some liquid (either from the refrigerator or from the condenser), and has no water jacket, but depends upon the latent heat of the liquid taken in with the vapour to reduce the temperature of the gas during compression. No attempt will be made to discuss the merits of either the wet or dry, single or double

acting systems, as that will come in under other papers. This paper will deal with the Boyle type of compressor, which is a vertical, single acting, dry compressor. All figures on its performance are such that for any other type of compressor they can be used with such modifications as are desired, due to the different volumetric efficiency of any other type of compressor.

With the Boyle type of compressor little or no superheating of the vapour above the temperature of the liquid in the refrigerator should occur before the vapour enters the compressor, and in all calculations to follow, the temperature of the vapour to the compressor has been assumed to be that of the liquid evaporating in the refrigerator.

In the steam engine cylinder condensation occurs, due to the fact that the cylinder walls are colder than the incoming steam, so in the compressor cylinder a like action takes place, only reversed. Whereas the walls of the steam engine cylinder took heat from the steam and partly condensed it, *the walls of the compressor cylinder give heat to the vapour, and so increase the bulk of a given weight of vapour, and prevent taking in a cylinder full of vapour at the density due to the temperature at which it enters the compressor cylinder.* For steam engines it has been pretty well demonstrated that for a given cylinder the cylinder condensation is proportional to the difference in temperature of inlet and outlet steam.

Let us apply this theory to the compressor to determine its volumetric efficiency.

The only complete published compressor tests known of by the author that were made by a competent disinterested authority are those given in Vol. XII., 1890,

of the transactions of the Am. Soc. Mech. Engrs., by Prof. J. E. Denton, on a 12in. × 30in. two-cylinder vertical, single-acting Boyle compressor, driven by a vertical simple Corliss engine. In these tests *the volumetric efficiency of the compressor*, which *is the ratio of the actual weight of ammonia pumped to the theoretical weight that should be pumped*, ranges from 73.5% to 84%. The volumetric efficiency of his exact test No. 1 at 28 lbs. suction and 151 lbs. condenser gauge pressure is 84%. For adiabatic compression of ammonia gas, that is compression without loss or gain of heat, Prof. Denton gives the following formula: $T_1 = T_0 \left(\frac{P_1}{P_0}\right)^{.24}$

T_1 = Absolute temperature of gas after compression = $t_1 + 460$

T_0 = Absolute temperature of gas to compressor = $t_0 + 460$

P_1 = Absolute pressure of gas after compression = $p_1 + 14.7$

P_0 = Absolute pressure of gas before compression = $p_0 + 14.7$

t_1 = Theoretical temperature of gas Fahr. degrees after compression.

t_0 = Temperature gas Fahr. degrees before compression.

p_1 = Gauge condenser pressure lbs. per square inch.

p_0 = Gauge refrigerator pressure lbs. per square inch.

By the use of this formula the theoretical temperature of gas at discharge from the compressor can be determined for any desired case. I have worked out a number of such cases from my own observation and from Denton's tests, and have derived the following

formula for the volumetric efficiency of the compressor. By using the volumetric efficiency and difference in temperature of gas after and before compression of Denton's No. 1 test above referred to (the temperature after compression being the theoretical temperature deduced from the above formula), we get the following constant: $t_1 - t_0 = 214^\circ$, volumetric efficiency $= .84$,

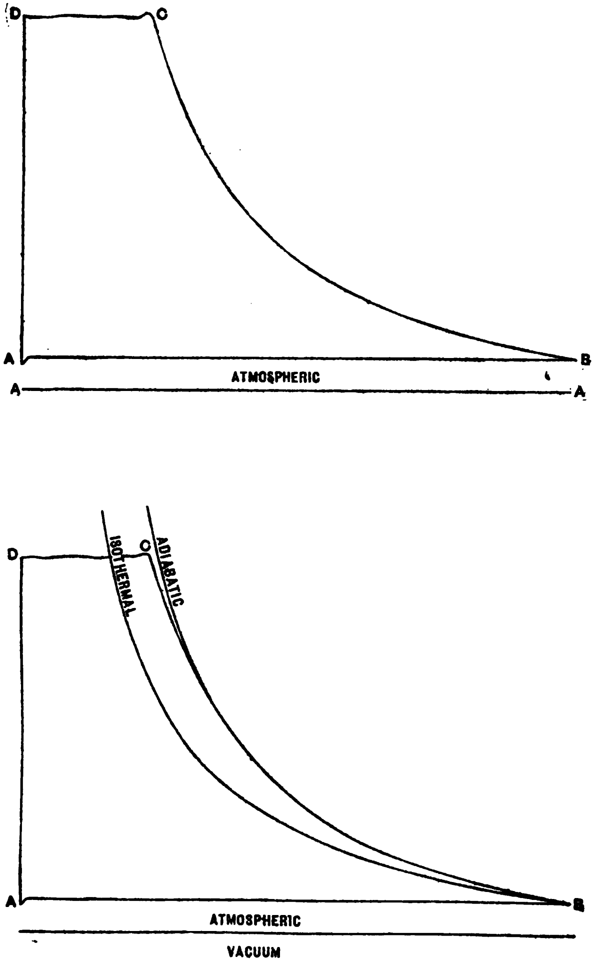
$$.84 = 1 - \frac{214}{x},\ x = 1330,\ E = 1 - \left(\frac{t_1 - t_0}{1330}\right)$$

where E is the volumetric efficiency of the compressor.

This formula for the volumetric efficiency of the compressor is the first formula of its kind that has ever been published, to my knowledge, and gives results that vary less than one per cent. from the volumetric efficiency actually found in Denton's eight tests before referred to.

From Denton's tests there is an average reduction of 57° below the theoretical for the temperature of the gas discharge from the compressor for his eight tests. The following table gives the theoretical discharge temperatures and volumetric efficiencies as worked out by the formula, and the actual cubic feet of displacement of compressor per ton of refrigerator per minute for the gauge condenser and suction pressures given:

Suction Pressures.		Condenser Pressures.		
		140	170	200
0	t_1	323°	358°	388°
	E	.76	.73	.71
	F	10.35	11.02	11.57
15	t_1	221°	254°	280°
	E	.83	.81	.79
	F	4.57	4.78	5.03
30	t_1	167°	192°	216°
	E	.87	.86	.84
	F	2.96	3.07	3.21


D
C
A
B
ATMOSPHERIC
A
A
D
C
ISOTHERMAL
ADIABATIC
A
B
ATMOSPHERIC
VACUUM

FIGS. 1 AND 2.

F = actual cubic feet of displacement per minute for compressor per ton of refrigeration, taking into account the volumetric efficiency of the compressor; in other words, F = the actual displacement in cubic feet per minute of the compressor to do one ton of refrigeration, and *not* the theoretical displacement in cubic feet per minute.

From this table at 15 lbs. suction and 170 lbs. condenser pressure the theoretical discharge temperature of gas from the compressor is 254°, the volumetric efficiency 81%, and the actual displacement of the compressor per minute per ton of refrigeration is 4.78 cubic feet.

Now, it is desired to know how much power it takes to drive the compressor, and to determine this we must take an indicator card from the compressor, and study it and find the horse-power of the compressor, and then determine the horse-power of the engine to drive the compressor.

Fig. 1 is an indicator card from a compressor, A B is the suction line, where distance B A is 15 lbs. to scale, B C is the compression line, C D is the discharge line, the distance D A being 170 lbs. to scale, and D A is the drop in pressure from the end of the discharge to the beginning of compression.

Let us now reproduce the card of fig. 1 in fig. 2, and draw on it the vacuum, adiabatic and isothermal lines. The vacuum line is 14.7 lbs. to scale below the atmospheric line; line B C will follow the adiabatic line if no heat were gained or lost by the gas during compression (outside of the heat given to the gas by compression) almost exactly as shown for almost its entire length. Line B C would follow the isothermal

line if the gas were compressed at a constant temperature.

For methods of construction of these curves and for determining the proper or improper operation of the compressor through their relations to the compression line b c (see "Indicating the Refrigeration Machine," by Voorhees, published by Nickerson and Collins Co., Chicago, U.S.A.) In this book simple constants are given whereby the adiabatic and isothermal lines can be easily and quickly laid out on any indicator card.

The isothermal compression of ammonia vapour is only theoretically possible, and is never approached with a dry compression machine in good order.

The adiabatic line is the curve that the compression line will almost exactly follow, and depart from it only as shown.

In hundreds of indicator cards I have taken from compressors in good condition I find that the above relation of the compression and adiabatic line is always maintained.

If the compression line b c does not almost exactly follow the adiabatic as shown there is something radically wrong with the compressor, as is fully explained in "Indicating the Refrigerating Machine."

Quoting from Professor Denton's tests, "No sensible change in the law of compression can be found; the (water) jacket may remove upward of half the heat represented by the power spent in compression, and yet not sensibly reduce the amount of the latter."

The Mean Effective Pressure of the indicator diagram is the average pressure the piston works against in compressing the gas, and, assuming an adiabatic compression, the mean effective pressure (hereafter called

M.E.P.) has been calculated for various suction and condenser pressures, and also the maximum M.E.P. and the percentage of stroke at which the discharge valve opens. The results of these calculations are given in the curves and formula on Plate I., which the author first published in *Ice and Refrigeration* in July, 1902. In a series of articles entitled "Analysing the Compressor" that appeared in *Ice and Refrigeration* in 1901 and 1902, the author deduced by the calculus the expression for the M.E.P. of the adiabatic ammonia indicator diagram, which was the first time that such an expression had been deduced and published. The value arrived at was

$$\text{M.E.P.} = 4.333\,p_0 \left(\frac{p_1}{p_0}\right)^{.231} - 1$$

p_0 = absolute suction pressure lbs. per square inch.
p_1 = absolute condenser pressure lbs. per square inch.

The value of the maximum M.E.P. occurs when

$$p_0 = \frac{p_1}{3.113}$$

The percentage of stroke as A B of fig. 1 or V_1 of curves Plate I.—during which gas is discharged into the condenser, is $v_1 = \left(\frac{p_0}{p_1}\right)^{\frac{1}{1.3}}$

Having the M.E.P. of the compressor indicator card, the compressor horse-power is readily found by the following formula:

C.H.P. = M.E.P. × F × .00437.
C.H.P. = compressor horse-power.
M.E.P. = M.E.P.
F = actual displacement of compressor in cubic feet per minute.

Denton's tests before mentioned give the *friction of the compressor and its engine as* 33⅓% of the compressor horse-power, or 25% of the engine horse-power. The author has found such a friction load well borne out in many tests he has made on several different makes of dry compressors, and, although many machine builders claim that their compressors require much less friction than this, the author believes that these results as obtained by Denton from a first-class machine in first class condition, and operated under favourable conditions, are such that he would hesitate to use a friction load of less quantity in any work that he was responsible for.

It is very misleading when one speaks of the friction of a compressor as, say, 20%, for it is left indefinite as to whether it is 20% of the engine horse-power or 20% of the compressor horse-power. As all calculations of the compressor must be deduced from the horse-power of compressor cylinder, it is much better to always refer to the friction of the compressor in terms of the compressor horse-power. The friction will be taken as 33⅓% of the compressor horse-power, or 25% of the engine horse-power in the work that follows. This will give the engine horse-power by the following formula:

$$\text{E.H.P.} = \frac{\text{C.H.P.}}{\cdot 75}$$

E.H.P. = Engine horse-power.

In the following table the mean effective pressure, compressor horse-power, and engine horse-power, where the volumetric efficiency of the compressor is, as before deduced, are given, and where the E.H.P. $= \dfrac{\text{C.H.P.}}{\cdot 75}$,

for the gauge condenser and suction pressures as given.

From this table for 15 lbs. suction and 170 lbs. condenser pressure, the mean effective pressure is 67, the

Suction Pressures.		Condenser Pressures.		
		140	170	200
0	M.E.P.	46.5	50.5	55.0
	C.H.P.	2.10	2.42	2.78
	E.H.P.	2.80	3.23	3.71
15	M.E.P.	59.5	67.0	74.5
	C.H.P.	1.19	1.40	1.64
	E.H.P.	1.59	1.87	2.19
30	M.E.P.	64.5	75.0	85.0
	C.H.P.	.83	1.00	1.19
	E.H.P.	1.11	1.33	1.59

compressor horse-power 1.40, and the engine horse-power 1.87 per ton of refrigeration.

Plate II. shows curves A for the indicated horse-power of engine, B for the actual cubic feet displacement per minute of compressor, and C for the volumetric efficiency of compressor; for single acting ammonia dry compression machines for gauge suction pressures from zero to 30 lbs. per square inch, and for gauge condenser pressures of 140, 170, and 200 lbs. per square inch, per ton of refrigeration per twenty-four hours.

For example, for 15 lbs. suction pressure and 170 lbs. condenser pressure for one ton of refrigeration per twenty-four hours, the indicated engine horse-power is found at a as 1.87; the cubic feet displacement per minute of compression is found at b, and is 4.78, which is net, account having been taken of the volumetric efficiency of the compressor. The volumetric efficiency of compressor is found at c, and is .81.

The indicated engine horse-power is in all cases the indicated compressor horse-power divided by .75, so the compressor horse-power in this case is 1.87 × .75, or 1.40.

Likewise any other compressor horse-power can be found.

Should the friction horse-power of the machine be different from that which has here been used, the engine horse-power can be found as follows: If the friction horse-power of the machine is such that E.H.P. equals C.H.P. divided by .85 in place of C.H.P. divided by .75, then the new engine horse-power is found by first obtaining the compressor horse-power as above, which is 1.40, and dividing 1.40 by .85, which equals 1.65 engine horse-power, and so the engine horse-power for any other case may be readily found.

If a different volumetric efficiency is to be used, say for 15 lbs. and 170 lbs. of 60% in place of 81% as here used, take the cubic feet displacement per ton from the curve and multiply by the volumetric efficiency from the curves, and divide by the desired volumetric efficiency, which, in this case, is 4.78 × .81 ÷ .60, which equals 6.46, which means that with 60% volumetric efficiency 6.46 cubic feet per minute must be displaced to give one ton of refrigeration.

But *the engine horse-power curves only hold good for the volumetric efficiency and displacements as given by the companion curves.* The E.H.P. curves can be used for any other ratio of E.H.P. and C.H.P., so long as the volumetric efficiency remains the same as shown by the curves. For a different volumetric efficiency, to find the horse-power requires the follow-

ing calculation: In the previous example, for 60% volumetric efficiency it was found that the displacement was 6.46 cubic feet per minute, and the compressor horse-power for this condition will be found by multiplying the displacement 6.46 by the mean effective pressure by the factor .00437. From the M.E.P. curves the M.E.P. for 15 lbs. suction and 170 lbs. condenser pressure is found to be 67, so the C.H.P. in this case is $6.46 \times 67 \times .00437$, or 1.89, and if the E.H.P. equal C.H.P. divided by .85, then the E.H.P. in this case is 1.89 divided by .85, which equals 2.23.

By the use of these curves (Plate II.) and rules, and knowing the volumetric efficiency and the value of C.H.P. divided by E.H.P., the capacity and power to operate any other type of compressor, whether it be single or double action, horizontal or vertical, wet or dry compression, can readily be found.

From observations in practice, and from theory as well, the *volumetric efficiency* of a dry compressor and its capacity to do refrigeration at a fixed speed and at fixed suction and condenser pressures is *greatest when the vapour from the refrigerator comes to the compressor with little or no superheat.* 30° superheat of the suction gas between the refrigerator and compressor reduces the capacity of the compressor 4%, and 100° superheat reduces the capacity 9%. This is because of the increased volume of a given weight of gas due to superheating, and because of the reduced volumetric efficiency due to the higher discharge temperature, more than overbalanced the refrigeration gained by using the specific heat of the cold vapour from the refrigerator to do refrigeration.

Although the temperature of the liquid to the expansion valve has been used as that due to its condenser pressure, this temperature in well-built condensers with liquid cooling apparatus (water cooled) will be reduced to within a few degrees of that of the coldest condensing water, as, for example, with 74° water the temperature of liquid to the expansion valve for 15 and 170 lbs. pressure can be 76° in place of 90°, whereby $90° - 76° = 14°$, and $14 \times 1 = 14$ B.T.U. per lb. liquid ammonia will be saved, which will increase the capacity of the compressor $\frac{14}{466} = 3\%$, and will reduce the horse-power per ton to 97% of that required if the liquid to the expansion valve were 90° in place of 76°.

The subject of condensers will not be taken up at length, except to say that of the many types, such as submerged, atmospheric, double pipe, etc., etc., for plants of any considerable size, the atmospheric type in a raised exposed location, roofed over and with louvred sides, is the best, and *for such condensers* the following rule is laid down—that *with liberal surface* they *will give* temperatures of condensation and *pressures corresponding thereto as per ammonia tables of 10° higher than the average temperature of the water to and from the condenser;* this for either compression or absorption machines with two gallons of water per ton per minute for all purposes and 1.8 gallons per ton per minute to the condenser. In all calculations for both compression and absorption machines in this paper two gallons of condenser water per ton of refrigeration per minute for all purposes has been used.

Great care should be taken that the expansion valves are below the liquid receiver of the condenser, and that the liquid lines from the liquid receiver do not have traps or pockets, and that the liquid lines shall not be run through places warmer than the condensing temperature, or, if so, that gas traps should be provided at the expansion valves to conduct gas formed in the liquid pipes back to the condenser, so that it does not pass through the expansion valve and so greatly reduce the capacity of the machine and greatly increase the power to operate it.

In general, always be sure by means of gauge glasses provided with automatic self-closing gauge cocks that nothing but liquid is going to the expansion valve, and, if possible, cool the liquid by means outside the machine, as by cooling water to a temperature lower than that due to the condensing pressure of the ammonia.

When apparently solid liquid at near its condensing temperature passes the expansion valve, the author believes that there is a large quantity of uncondensed gas in this liquid not optically visible but contained in the space between the liquid molecules which accounts for the poor results often obtained with high condensing pressures and temperatures of liquid to the expansion valve very near the condensing temperature of the ammonia.

MULTIPLE EFFECT COMPRESSOR (Voohees' Patent).

A simple form of the multiple effect compressor is shown in fig. 3. The multiple effect compressor takes in gas from two or more refrigerators at two or more

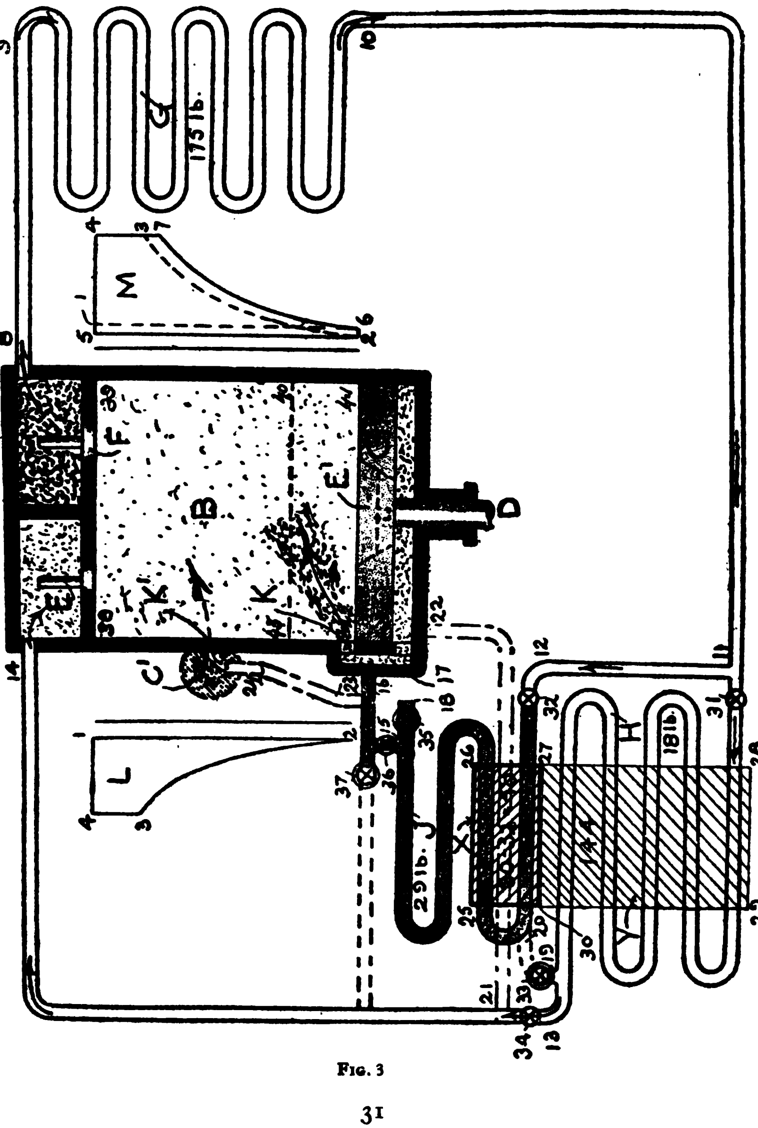


Fig. 3

different suction pressures and temperatures on the same suction stroke of the compressor. The suction gas of the higher pressure helps to compress the lower suction pressure gas. The multiple effect compressor is thus enabled to have a capacity equal to or greater than that due to the highest suction pressure it uses, while it does part of its refrigeration at a low suction pressure, and requires less power than would two compressors of the ordinary design doing the same amount of refrigeration at the two suction pressures, or than would be required if all the refrigeration were done with a larger compressor at the lowest suction pressure.

As two or more plains of temperature are or can be used in most all refrigerating or ice making plants, the adaptability of the multiple effect compressor will be apparent. For example, a multiple effect compressor operating at 15 and 30 lbs. gauge suction pressures will do refrigeration at the increased rate of the ratio of the absolute suction pressures minus one or $\frac{30 + 15}{15 + 15} - 1 = \frac{45}{30} - 1 = 1.5 - 1 = .5$, or 50% more of refrigeration. In fig. 3 liquid ammonia flows from condenser G through pipe 10, 11, 12 to expansion valves 31 and 32.

In expansion coil H the gas expands at, say, 18 lbs. gauge back pressure, and in expansion coil J at, say, 29 lbs. gauge back pressure. The vapour from coil H flows to the compressor A through pipe 13, 14, and enters the compressor through suction valve E in the usual way, filling the cylinder with 18 lbs. pressure gas. When the piston C has reached the position shown in the figure, it uncovers the port K, and then the 29 lbs.

pressure gas from expansion coil J rushes into the cylinder through pipe 36, 16, and port K and compresses the cylinder full of 18 lbs. pressure gas to 29 lbs. pressure, and so makes room for additional 29 lbs. pressure gas; then the piston covers port K on its return stroke and compresses the cylinder full of 29 lbs. pressure gas and discharges it into the condenser G in the usual manner.

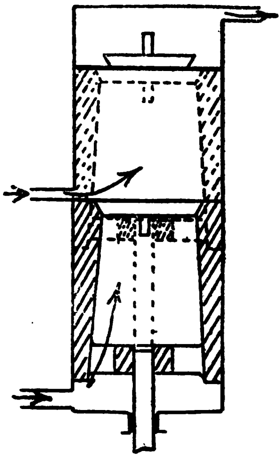


FIG. 3a.

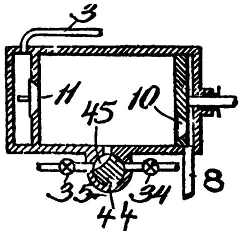


FIG. 3b.

Any compressor, whether horizontal or vertical, single or double acting, wet or dry, can readily be made into a multiple effect compressor. Full description and tests of the multiple effect compressor are given in a paper by the author in Vol. 2, 1906, "Transactions of the American Society of Refrigerating Engineers," and in January, 1907, number of *Ice and Refrigeration*, also in the March and July numbers of *Ice and Refrigeration*, 1905.

Fig. 3a is a valve in the piston type multiple effect compressor; fig. 3b is a valve in the piston type multiple effect compressor with a positively operated high suction pressure valve; fig. 3c is a double-acting multiple effect compressor; fig. 3d is a double-acting

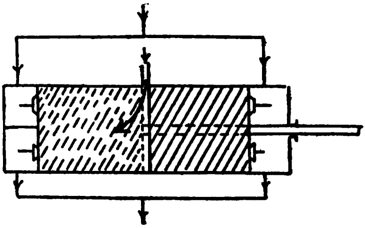


Fig. 3c.

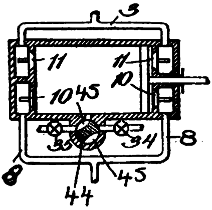


Fig. 3d.

multiple effect compressor with a positively operated high suction pressure valve. In fig. 3a the piston is a little longer than the stroke; in fig. 3c the piston is the same length as the stroke, and the cylinder twice the length of the stroke plus the width of the high suction pressure ports. Machines as large as

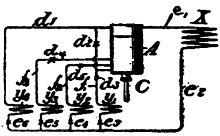


Fig. 3e.

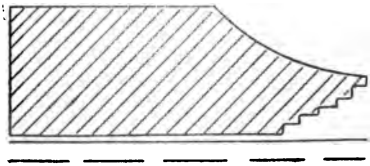


Fig. 3f.

15in. by 32in. have been fitted up for multiple effect compressors, and in one test made by the author on a 14in. by 32in. multiple effect compressor at zero and 7 lbs. suction pressure he actually obtained 89% increased capacity for 32% less power than if two compressors operating at zero and 7 lb. gauge suction

pressure had been used, or 68% more power than that used by the multiple effect compressor would have been required to do the same quantity of refrigeration with a larger non-multiple effect compressor, all at zero pounds suction pressure (gauge). Fig. 3e is a diagrammatic view of a multiple effect compressor with five effects, and fig. 3f is an indicator card from such a five-effect compressor. A multiple effect compressor with six effects will do 400% more refrigeration at the same speed and displacement as it would do if making plate ice at zero pounds gauge suction pressure if it did refrigeration in the following stages.

	Gauge Suction Pressure.
Plate ice	0
Can ice	15
Brewery	25
Forecoolers	35
Baudelots	45
Air coolers for blast furnaces, or for cooling living rooms	60

Fig. 3g shows the relation of indicator cards from a common and a multiple effect compressor, and are thus explained:

In analysing the action of the compressor by the use of diagram 1, A A is the atmospheric line, V V is the vacuum line, Vb_1 the absolute suction pressure in the compressor cylinder at the end of the suction stroke, $a_1\,b_1$ is the suction line, $b_1\,c_1$ the compression curve, $c_1\,d_1$ the discharge line, and $d_1\,a_1$ the drop in pressure from the end of discharge to the beginning of the next suction stroke.

If the temperature of the gas at discharge is constant, then *for a given discharge pressure* the length of line $c_1\,d_1$ is proportional to the weight of gas pumped, and

therefore a measure of the refrigeration done by the compressor. If line $c_1 d_1$ were twice as long the compressor would be doing twice as much refrigeration, and if the line $c_1 d_1$ were half as long the compressor would be doing half as much refrigeration. With the suction gas at constant temperature, if the *absolute* suction pressure Vb_1 were twice as much the compressor would be doing twice as much refrigeration, and if half as much then the compressor would be doing half as much refrigeration.

Let indicator diagram 2 be taken from a compressor of half the displacement and at the same discharge pressure of that of diagram 1. Here the suction pressure Vb_2 and the discharge line $c_2 d_2$ are twice as long as the suction pressure Vb_1 and the discharge line $c_1 d_1$ of indicator diagram 1, so that this compressor, of one-half the displacement of that of diagram 1, pumps gas twice as dense, and so does just as much refrigeration, as the compressor of twice the displacement.

Place diagram 2 on diagram 1, as shown in indicator diagram 3, so that vacuum lines V V and discharge points d_1 and d_2 coincide. As the suction pressure Vb_2 is greater than the suction pressure Vb_1, it will be evident that if a compressor is properly designed gas at the higher suction pressure will flow into the cylinder that is already full of low pressure gas until the lower pressure gas has made room for and been compressed by the high suction pressure gas to its high pressure. A cylinder so designed and operated I have called a Multiple Effect Compressor (hereafter referred to as M.E.C.). Its

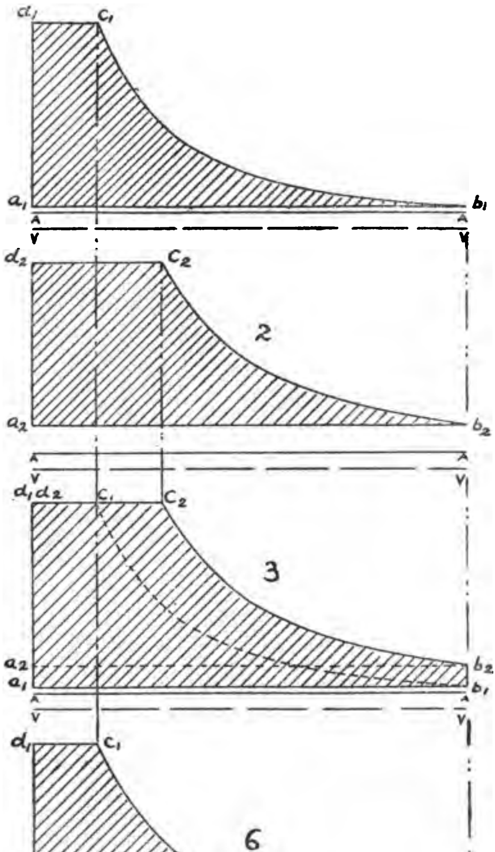


FIG. 3g.

etical indicator diagram is the outside bounding line of diagram 3 and its action is — suction of low pressure gas on line a_1 b_1, inflow of high suction pressure gas at b_1, and compression of low pressure gas thereby to b_2, without the aid of energy from the steam engine; compression of the gas from b_2 to c_2 and its discharge to d_1 d_2 and drop of pressure to the beginning of the next suction stroke a_1. The M.E.C. diagram 3 shows that at the same speed and with the same displacement as the compressor that made diagram 1 the M.E.C. did all the refrigeration done by the diagram 1 compressor plus that which was done by the diagram 2 compressor, so that it produced 100% more refrigeration than did the compressor of like displacement operated at the same speed and at the same low back pressure.

ABSORPTION MACHINES.

The absorption machine has a refrigerator and condenser similar to that of the compression machine, uses the same weights of ammonia per ton refrigeration for the same refrigerator and condenser pressures, and, broadly, differs only in that the *absorber takes the place of the suction stroke of the compressor and the still or generator takes the place of the compression stroke of the compressor.* In its simplest form, ammonia vapour from the refrigerator is absorbed by weak liquor (a combination of water and ammonia, called aqua ammonia) in the absorber, the heat of absorption being taken out by cooling water. The weak liquor then becomes strong liquor, and is pumped by a liquor pump from the absorber into the generator, wherein the ammonia is distilled from the strong

liquor by the action of heat, as by steam coils, and the resultant weak liquor returns to the absorber to repeat the cycle, and the distilled ammonia vapour from the generator is condensed in the condenser, and passed through the expansion valve to the refrigerator, and there evaporated in doing refrigeration, and is returned as vapour to the absorber and the cycle repeated.

In modern absorption machines the following additional apparatus is used: An exchanger, to give part of the heat of the hot weak liquor to the cold strong liquor; and a rectifier, to remove the water vapour from the vapour from the generator before it is condensed in the condenser. Everything before said about refrigerators, condensers, liquid lines, etc., refers to the absorption machine as well as to the compression machine.

It is often erroneously stated that an absorption machine requires more and colder water than does a compression machine. Later, figures will be given for an absorption machine operated with the same quantity of water at the same temperature as a compression machine, and a comparison of results and economies and efficiencies. It is true that the use of an excess of water will give better results with an absorption machine than the same quantity of water will with a compression machine, for such an increase of water will not increase the capacity of a compression machine or its efficiency as much as it will increase the capacity and efficiency of an absorption machine.

One of the greatest drawbacks to the absorption machine is the improper proportioning or operation of its different parts.

A compression machine with a given condenser and refrigerator, and with liquid to the expansion valve, must, if the compressor is in good order, give positive results nearly in proportion to the speed of the compressor. For every suction stroke made a discharge stroke must also be made, while in the absorption machine there is no positive action of the various pieces of apparatus, and the good or bad work of the designer of the absorption machine and the condition of each piece of apparatus and the personal equation of the operating engineer, all make very radical differences in the economy, capacity, and efficiency of the absorption machine, for here the old saying that a chain is no stronger than its weakest link is well borne out. A poorly constructed or operated absorber exchanger or generator or any part of the absorption machine apparatus will change the operation of the entire combined apparatus approximately to the low capacity, economy, or efficiency of the poorest of any of these parts.

In general, the following fundamental rules can be laid down for an absorption machine to give best results. The generator should have ample liquid evaporating surface to make dry gas. This is on the same principle that vertical steam boilers do not give as dry steam as do horizontal steam boilers. As the partial pressure of the water vapour in the gas to the rectifier bears a direct relation to its temperature, and also a direct relation to the weight of water vapour in the combined water and ammonia vapour, it is essential that the temperature of the gas to the rectifier should be as low as possible, so that the least possible water

vapour goes to the rectifier. The drip liquor returned to the generator from the rectifier should be as hot as possible, for otherwise it contains too much ammonia. The temperature of gas from the rectifier to the condenser should not be over from 10° to 50° hotter than the condensing temperatures of the gas, for otherwise it will carry over water vapour to the condenser. The gas from the rectifier to the condenser should not be too near its condensing temperature, as if so too much ammonia will be condensed in the rectifier. The exchanger should exchange upwards of 90% of the heat of the hot weak liquor to the cold strong liquor. The quantity of strong liquor pumped per lb. of anhydrous ammonia circulated in the refrigerator should be such that the combined heat required to heat the strong liquor to the temperature of the weak liquor plus that to re-evaporate the condensed water vapour and ammonia (drip liquor) returned from the rectifier is a minimum. For well proportioned machines this is found from both practice and theory to be between seven and eight; that is, seven to eight lbs. of strong liquor are circulated for every lb. of anhydrous ammonia circulated in the refrigerator. The strength of the strong liquor depends upon its temperature and pressure, as it leaves the absorber, and the strength of the weak liquor depends upon its temperature and pressure as it leaves the generator.

In the results to be given the author has used the data from upwards of hundreds of tests made by him of absorption machines of various makes. In order to properly work up such results the author has had to build up data that have taken him years of practice, experiment, and research to perfect.

The principal data required are as follows: The relation of pressure and temperature of various strengths of aqua ammonia. The specific gravity of aqua ammonia at different temperatures and pressures. The co-efficient of expansion of various strengths of aqua ammonia at different temperatures. The partial pressure of water vapour in ammonia vapour at various temperatures and pressures. The heat of absorption and disassociation of ammonia in water at various differences of percentages of ammonia in water. The efficiency of pipe surface in generators, absorbers, exchangers, rectifiers, etc. Formula for the operation of generators, absorbers, exchangers, rectifiers, etc., for fixed conditions.

One of the most difficult matters to adjust to my satisfaction was to obtain the true percentage of ammonia in aqua ammonia. The usual form of floating hydrometer is of absolutely no value for any except the crudest practical determinations. The personal equation of the observer and the great variation of the percentage of ammonia for a small variation in the hydrometer reading is sufficient to defeat the possibility of good results. For accurate results, the author used the Westphal Balance, and found that even then no correct results could be obtained with the temperature of the aqua ammonia anywhere near its boiling point. The 60° Fahrenheit temperature standard for hydrometers usually used is altogether too high, as many strengths of aqua ammonia cannot exist at atmospheric pressures at 60° Fahr., so that the author established for his work a standard temperature of —30° Fahr. for all hydrometer readings, to which temperature all liquors are reduced when being examined by the

Westphal Balance. The use of the Baumé scale is of no value for aqua ammonia, and not only forms an unnecessary step that confuses operating engineers, but also is quite misleading, for the relation of Baumé degrees and percentages of ammonia in a solution is arbitrary, and requires the use of tables to transform the results to percentages of ammonia that are quite unnecessary and open to error. For example, 10% aqua equals 16° Baumé; 20% aqua equals 22° Baumé; 30% aqua equals 27° Baumé, approximately. If the difference between strong and weak liquor were to be 10%, then with 10% and 20% aqua there would be a difference of 6° Baumé, while if the difference were to be 10% with 20% and 30% aqua, the difference would be only 5° Baumé. *If a floating hydrometer is used at all, it should be marked in percentages of ammonia and not in Baumé degrees.*

In the data to follow exchangers have been used that exchange 90% of the available heat of the hot weak liquor to the cold strong liquor, and it is assumed that the liquor pump pumps 8 lbs. of strong liquor to 1 lb. of anhydrous ammonia evaporated in the refrigerator, and that the surfaces of absorber, rectifier, analyser, and generator are what one would find in a first-class, up-to-date, modern absorption machine, and when a special generator is referred to it means one with a larger steam coil surface than is commonly used for standard machines, and one which is practical as to first cost.

In fig. 4 an absorption machine is shown operated at 15 lbs. suction and 170 lbs. condenser gauge pressures, and using two gallons of condensing water per minute per ton of refrigeration for the condenser,

rectifier, and absorber, and doing one ton of refrigeration per twenty-four hours. Here 30.9 lbs. of dry steam not superheated at 38 lbs. gauge pressure is condensed in the coils of the generator in evaporating 32% strong liquor to 22.3% weak liquor; 3.01 lbs. of weak liquor leaves the generator at 22.3% of ammonia at 264°, and gives up its heat in the exchanger, so that it flows to the absorber at 131°. In many machines using wet absorbers (a dry absorber is shown here), a weak liquor cooler is used, as shown, where the weak liquor from the exchanger is further cooled by cooling water, but as the author does not consider the weak liquor cooler necessary, he has not used it in the calculations to follow or in the operation of the machine shown in fig. 4. The weak liquor at 131° and 22.3% now sprays into the absorber, and absorbs .43 lb. of ammonia vapour from the brine cooler, and becomes 3.44 lbs. of strong liquor of 32% and 111°. The absorber here shown is the dry absorber, and has the greatest efficiency of any absorber of which the author knows.

Another form of absorber largely used is the wet absorber, in which the absorber is full of liquor, the weak liquor coming in at the top and the strong liquor going out at the bottom, while the vapour comes in at the bottom and bubbles up through the liquor until it is absorbed therein. Dry absorbers seem to give higher absorption pressures than wet ones, but this is due mainly to improperly connecting the gauge to the wet absorber; it should be connected at the bottom, so it will have the liquid head due to the height of the liquid in the absorber, as it will then only represent the total pressure at which the temperature and pressure

govern the strength of the strong liquor. The 3.44 lbs. of strong liquor (32% and 111°) is pumped by a liquor pump from the absorber through the exchanger into the analyser. In passing through the exchanger the strong liquor is heated and partly evaporated, so that it enters the analyser at a temperature of 224° as liquor of a little less than 32% ammonia and some gas, in all a weight of 3.44 lbs. Ammonia and water vapour are evaporated from the liquid surface in the generator, and flow up in contact with the shelves of the analyser, over which the combined strong and drip liquors are flowing down to the generator, the upward flowing gas being cooled and some water vapour and ammonia condensed, and the downward flowing liquor being heated and some water vapour and ammonia gas evaporated. The combined ammonia and water vapours leave the generator as .4982 lb. of the mixture at 234° and 91% of ammonia vapour.

This combined water and ammonia vapour passes up through the rectifier, and is cooled to 110°, and in so doing condenses the water vapour, which absorbs a quantity of ammonia due to its temperature, and this condensation (called drip liquor) flows in a counter current down the rectifier in contact with the upward flowing hot gas, and finally flows out through the trap into the top of the analyser as .0682 lb. of drip liquor at 214° and containing 34% of ammonia. The cooling water flowing over the rectifier takes out the heat. .43 lb. of ammonia gas, now practically pure anhydrous ammonia, flows at 110° to the condenser, wherein it is cooled to 90° and condensed to a liquid at 90°, the heat of con-

densation being taken out by the cooling coils. .43 lb. of anhydrous ammonia at 90° now flows through the expansion valve to the brine cooler, where its temperature is reduced to zero degrees, and it boils as a liquid in the brine cooler in cooling three gallons of brine per minute from 12° to 3°, and passes off as a vapour at zero degrees, to be absorbed in the absorber, and the cycle repeated. Two gallons per minute of cooling water at 73° flows through the condenser, where it is heated to 86°, and .2 gallon of the 86° water goes to the rectifier, where it is heated to 142°, and 1.8 gallons at 86° goes through the absorber, where it is heated to 110°.

Assuming the mechanical efficiency of the liquor pump to be .75, then the horse-power used in a liquor pump engine is .064 h.p., and if it were driven by a simple Corliss engine it would have a cylinder condensation of .46 lb. of steam per hour, and would deliver the balance of its exhaust steam to the generator at 38 lbs. gauge pressure to make a part of 30.9 lbs. of steam used in the generator, so that the total steam used in the generator and liquid pump would be 31.36 lbs. per ton of refrigeration per twenty-four hours. The steam in the generator was used to make up the difference in the heat not given back to the strong liquor by the weak liquor, to reduce the strong liquor to the weak liquor, and to re-evaporate the drip liquor. The heat taken out of the system by the condenser (220 B.T.U.) plus the heat taken out by the absorber (383 B.T.U.) plus the heat taken out by the rectifier (93 B.T.U.), or a total of 696 B.T.U., equals the heat given to the system of 496 B.T.U. in the generator plus 200 B.T.U. in the brine cooler, or

696 B.T.U. The following table is with two gallons of condensing water per minute for condenser, absorber, and rectifier to do one ton of refrigeration per twenty-four hours. In this table the steam from the liquor pump is not exhausted into the generator, and the liquor pump is supposed to take steam at the rate of 30 lbs. per horse-power of engine driving the liquor pump per hour. The pounds of strong liquor to pounds of anhydrous ammonia evaporated in the refrigerator is eight to one.

Suction Pressures.		Condenser Pressures.		
		140	170	200
0	Sl	24%	22%	18%
	Wl	13.13%	10.85%	6.28%
	SG	30.1	41.3	48.7
	SL	1.7	2.1	2.4
	SGL	31.8	43.4	51.1
15	Sl	35%	32%	28%
	Wl	25.75%	22.3%	17.7%
	SG	27.9	30.9	34.1
	SL	1.6	1.9	2.3
	SGL	29.5	32.8	36.4
30	Sl	42%	38%	36%
	Wl	33.70	29.15%	26.9%
	SG	22.9	26.2	27.9
	SL	1.4	1.8	2.2
	SGL	24.3	28.0	30.1

Sl = strong liquor. Wl = weak liquor. SG = lbs. of steam per ton of refrigeration per hour for the generator. SL = lbs. of steam per ton of refrigeration per hour for the liquor pump. SGL = lbs. of steam per ton of refrigeration per hour for the generator and liquor pump. The pressures given are in lbs. per square inch gauge.

Referring again to fig. 4, the compression machine here takes .43 lb. of vapour from the brine cooler at

15 lbs. and zero degrees, and compresses it to 170 lbs. at 224°, and discharges it into the condenser, where it is cooled to 90° and condensed to a liquid at 90°, the compressor adding 59 B.T.U. to the gas due to the heat of compression, the water jacket taking out 24% of the heat of compression, or 14 B.T.U., leaving 215 + 45 = 260 B.T.U. to be taken out by the condenser. The heat balance for the compression machine has not been made up here, as there are certain elements relating to the friction of compressor and latent heats of evaporation and condensation that will not be discussed in this paper; and the same questions of latent heats of condensation and evaporation affect the approximate heat balance shown for the absorption machine, so that that heat balance must not be considered correct, but only as indicating how such a heat balance may be made up from the heats received and rejected by the ammonia in its various stages in the cycle.

The engine in fig. 4 that drives the compressor is a simple Corliss engine, and exhausts into the atmosphere (or it could exhaust into the generator). It will take 28 lbs. of steam per horsepower per hour, and so will for 1.87 h.p. use 1.87 × 28 = 52.4 lbs. steam per hour per ton of refrigeration, exhausting into the atmosphere.

The relative capacities of compression and absorption machines taken at the suction and condenser gauge pressures given are shown in the following table, where C equals the relative compressor capacity and A the relative absorption machine capacity, being referred to a capacity of 1 at 15 lbs. suction and 170 lbs. gauge condenser pressure.

The relative surfaces of condensers are in about the ratio of one for the compression to .84 for the absorption machine. The condenser temperatures used are 80°, 90°, and 100° for 140, 170, and 200 lbs. gauge

Suction Pressures.		Condenser Pressures. 140	170	200
0	C A	.46 .97	.43 .94	.41 .91
15	C A	1.05 1.03	1.0 1.0	.95 .97
30	C A	1.62 1.09	1.56 1.05	1.49 1.03

condenser pressure, and the temperatures of water to condensers used are 63°, 73°, and 83° for two gallons of water per ton of refrigeration per minute for 140, 170, and 200 lbs. condenser gauge pressures.

The same refrigerator or condenser cannot be used for an absorption machine and for a compression machine at the same time or intermittently, for the oil from the compression machine will become carbonised in the generator and create non-condensable gases in the condenser.

From a careful study of a number of engine tests the following figures have been used for steam per indicated engine horse-power per hour (I.E.H.P.) for Corliss engines, and for percentage of cylinder condensation. The percentage of cylinder condensation is included in the steam used for horse-power per hour, so that the figures for steam used for horse-power per hour are total. These figures are somewhat low, and will be difficult to realise with a slow running compressor except under full load and with most careful adjustment of steam and exhaust valves.

In the compression machine, the engine, if not operated at full load, will require about 10% more steam for 50% overload, and about 30% more steam for 50% underload, while the steam consumption per ton of refrigeration for an absorption machine will be less for underload and very little more for overload.

Type of Engine.	Lbs. of Steam per I.E.H.P. per Hour, including Cylinder Condensation.	Cylinder Condensation.
Simple non-condensing	28	26%
Compound non-condensing	20	18%
Triple expansion non-condensing ..	17	12%
Compound condensing	15	20%
Triple expansion condensing	13	12%

The steam per ton of refrigeration per hour for compression and absorption machines has been calculated and the results given in Plate III. Here the compressor is operated first by a simple non-condensing Corliss engine, and, second, by a compound condensing Corliss engine. The liquor pump of the absorption machine does not exhaust into the generator, and is supposed to take 30 lbs. of steam per indicated horsepower per hour of the engine driving it. As none of the specific heat of the cold vapour from the refrigerator was used to do refrigeration in the compression machine system, so also none of that specific heat has been used in the absorption machine results. But this vapour can be used to advantage in the absorption machine, and will, if superheated 30°, increase the efficiency and capacity about 3½%, but let us call this equal to a possible loss of efficiency and capacity

due to a little water vapour carried over from the rectifier to the condenser.

It will be seen from these curves that the economy of the absorption machine is much better for all conditions than that of a simple non-condensing engine driven compressor, and that at between 8 and 10 lbs. gauge suction pressure that the economy of absorption and the compound condensing engine driven compressor is the same, and that at suction pressures above 8 to 10 lbs. gauge the economy of the compound condensing engine driven compressor exceeds that of the absorption machine, and that the economy of the absorption machine exceeds that of the compound condensing engine driven compressor at pressures below 8 to 10 lbs. gauge suction.

The three curves for the steam per ton of refrigeration per hour of the absorption machine may look quite simple, but it required the data from hundreds of absorption machine tests, and then nearly two weeks of constant application, to produce these curves, which, in so far as is known, are the first of the kind ever published.

The following table gives the steam consumption for simple non-condensing and for compound condensing engine driven compressors and for absorption machines with liquor pump not exhausting into generator, at the suction and condenser pressures (gauge) given:

SC = Simple non-condensing engine driven compressor.

CC = Compound condensing engine driven compressor.

A = Absorption machine, liquor pump not exhausting into generator, total steam for generator and liquor pump.

Suction Pressures.		Condenser Pressures.		
		140	170	200
0	SC	78.3	90.5	104.0
	CC	42.0	48.4	55.6
	A	31.8	43.4	51.1
15	SC	44.5	52.5	61.4
	A	20.5	32.8	36.4
	CC	23.8	28.0	32.7
30	SC	31.1	37.2	44.5
	A	24.3	28.0	30.1
	CC	16.6	19.0	23.9

To produce one ton of refrigeration at 8½ lbs. suction and 170 lbs. gauge condenser pressure, about 3.5 times as many heat units are actually used by an absorption machine as by a compression machine (compound condensing engine driven), but, owing to the low efficiency of the steam engine, due to the heat wasted in exhaust and by cylinder condensation, the actual weight of steam used per ton of refrigeration per hour is the same for both absorption and compound condensing engine driven compressor.

What may be of most general interest in a paper of this kind is the ice-making question, and as that question will be dealt with in other papers, we will here only apply the refrigeration as has been calculated for the production of plate ice, and in the figures to be given it has been assumed that the generator of the absorption machine is a special generator (that is, one having considerably more steam coil surface per ton of refrigeration than is used in the standard makes of absorption machines).

It is assumed that 1.6 tons of refrigeration are required to produce one net ton of plate ice as harvested, and that .4 horse-power per net ton of ice will be required to operate all auxiliary machinery, including everything, such as electric lights, water pumps, brine pumps, etc. The following tons of ice per ton of coal (2,000 lbs. equal one ton) have been deduced for boilers evaporating 8.5 lbs. of water per lb. of coal, with 73° water (to be made into ice) to the fore-cooler, and 15 lbs. suction and 170 lbs. gauge condenser pressure and 30 lbs. suction gauge pressure for the high suction pressure of the multiple effect compressor. The figures given are only 80% of what one would obtain with best operation of plant at 8½ lbs. water evaporated per lb. of coal in the boiler. In other words, the figures given below can, with best operation, be increased to an amount arrived at by dividing the figures as given by .80. *The figures given are such as the author would be willing to guarantee for any plant erected entirely according to his design and under his supervision.*

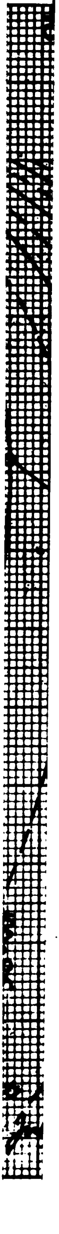
Where the absorption machine is used in combination with the compression machine or with the multiple effect compression machine, all of the exhaust steam from the engine driving the compressor, the liquor pump, and the auxiliary machinery is exhausted into the generator of the absorption machine, and all apparatus, such as compressor, liquor pump, and auxiliaries of every kind, are driven by one engine of the type designated. The percentage of the ice made by the absorption machine in combination with a compression or multiple effect compression machine is also given:

REFRIGERATING MACHINES.

Type of Refrigerating Machine and Type of Engine.	Net Tons of Ice per Ton of Coal.	% of Ice by Absorption Machine.
Compression, simple Corliss engine, non-condensing	6.1	—
Compression, simple Corliss engine, non-condensing, multiple effect	7.6	—
Compression, compound Corliss engine, non-condensing	8.3	—
Compression, triple expansion, Corliss engine, non-condensing	9.7	—
Absorption, liquor pump and auxiliaries, not exhausting into generator, simple non-condensing engine	10.0	—
Compression, compound, non-condensing engine, multiple effect	10.8	—
Compression, compound condensing engine	11.2	—
Compression, triple expansion engine, non-condensing, multiple effect	12.8	—
Compression, triple expansion engine, condensing	12.8	—
Absorption, liquor pump and all auxiliaries exhausting into generator, simple Corliss engine, non-condensing	13.3	—
Compression and absorption, simple Corliss engine, non-condensing	13.4	67.5
Compression, compound Corliss engine, condensing, multiple effect	14.5	—
Compression and absorption, simple engine, non-condensing, multiple effect	14.6	61.5
Compression and absorption, compound engine, non-condensing	16.0	60.8
Compression, triple-expansion condensing engine, multiple effect	16.5	—
Compression and absorption, triple-expansion non-condensing engine	17.6	58.0
Compression and absorption, compound non-condensing engine, multiple effect	17.6	55.0
Compression and absorption, triple-expansion non-condensing engine, multiple effect	19.5	51.8

It is a pleasure to note that the highest rate of ice per ton of coal is obtained by a combination of the compression, multiple effect compression, and the absorption systems.

Ca



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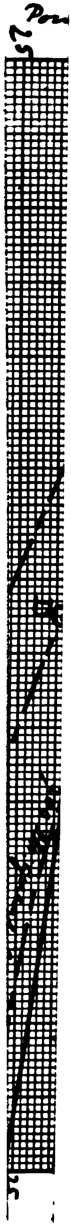

Tables of Temperatures, Proportions and Capacities.

Reprinted from "Ice and Cold Storage Trades' Directory, 1909"

PROPERTIES OF SATURATED STEAM AT PRESSURE FROM 1 LB. TO 200 LBS. ON THE SQUARE INCH.

Pressure Absolute.		Heat in Degrees Fahr.					Volume that of an equal weight of Water at its greatest density being 1.	Weight of one cubic foot in Decimals of a pound.	Specific Gravity, the atmosphere at 32° being 1.
In lbs. per sq. in.	In inches of Mercury at 32°	Temperature.	Dif. pr. lb.	Latent Heat.	Total Heat.	Dif. pr. lb.			
1	2·0375	102·	....	1,043·05	1,145·05	....	20,890	·0029	·037
5	10·1875	162·37	9·26	1,001·9	1,163·46	2·82	4,627	·0135	·167
10	20·375	193·29	4·93	979·60	1,172·80	1·50	2,429	·0257	·318
15	30·5625	213·07	3·47	965·85	1,178·92	1·05	1,669	·0373	·468
20	40·75	228·	2·6	955·5	1,183·5	·8	1,280	·0487	·604
25	50·9375	240·2	2·3	947·	1,187·2	·7	1,042	·0598	·742
30	61·125	250·4	2·	939·9	1,190·3	·6	881	·0707	·877
35	71·3125	259·3	1·7	933·7	1,193·	·5	764	·0815	1·012
40	81·5	267·3	1·5	928·1	1,195·4	·4	676	·0921	1·142
45	91·6875	274·4	1·4	923·2	1,197·6	·4	608	·1025	1·272
50	101·875	281·	1·3	918·6	1,199·6	·4	552	·1129	1·402
55	112·0625	287·1	1·2	914·4	1,201·5	·4	506	·1232	1·529
60	122·25	292·7	1·1	910·5	1,203·2	·3	467	·1335	1·654
65	132·4375	298·	1·1	906·8	1,204·8	·3	434	·1436	1·779
70	142·625	302·9	1·	903·4	1,206·3	·3	406	·1536	1·904
75	152·8125	307·5	·9	900·3	1,207·8	·3	381	·1636	2·029
80	163·	312·	·9	897·1	1,209·1	·2	359	·1736	2·151
85	173·1875	316·1	·8	894·3	1,210·4	·3	340	·1833	2·271
90	183·375	320·2	·8	891·4	1,211·6	·2	323	·1930	2·391
95	193·5625	324·1	·8	888·7	1,212·8	·3	307	·2030	2·511
100	203·75	327·8	·7	886·1	1,213·9	·2	293	·2127	2·631
105	213·9375	331·3	·7	883·7	1,215·0	·2	281	·2224	2·751
110	224·125	334·6	·6	881·4	1,216·0	·2	269	·2319	2·871
115	234·3125	338·	·6	879·	1,217·0	·2	259	·2410	2·990
120	244·5	341·1	·6	876·9	1,218·0	·2	249	·2503	3·105
125	254·6875	344·2	·6	874·7	1,218·9	·2	239	·2596	3·227
130	264·875	347·2	·6	872·6	1,219·8	·2	231	·2693	3·347
135	275·0625	350·	·5	870·7	1,220·7	·1	223	·2788	3·467
140	285·25	352·9	·6	868·6	1,221·5	·1	216	·2883	3·582
145	295·4375	355·6	·6	866·8	1,222·4	·2	209	·2978	3·697
150	305·625	358·3	·5	864·9	1,223·2	·2	203	·3073	3·809
155	315·8125	360·9	·5	863·1	1,224·	·2	196	·3168	3·927
160	326·	363·4	·5	861·4	1,224·8	·2	191	·3263	4·042
165	336·1875	365·9	·5	859·7	1,225·6	·2	186	·3353	4·157
170	346·375	368·2	·4	858·1	1,226·3	·2	181	·3443	4·270
175	356·5625	370·6	·5	856·4	1,227·	·1	176	·3533	4·383
180	366·75	372·9	·4	854·8	1,227·7	·1	172	·3623	4·495
185	376·9375	375·3	·5	853·1	1,228·4	·1	168	·3713	4·607
190	387·125	377·5	·4	851·6	1,229·1	·1	164	·3800	4·720
195	396·3125	379·7	·4	850·1	1,229·8	·2	160	·3888	4·832
200	407·5	381·7	·3	848·6	1,230·3	·1	157	·3973	4·945

F

FREEZING TIMES FOR DIFFERENT TEMPERATURES AND THICKNESSES OF CAN ICE.

(From the "Compend of Mechanical Refrigeration.")

Thickness.	1in.	2in.	3in.	4in.	5in.	6in.	7in.	8in	9in.	10in.	11in.	12in.
Temperature 10°	0·32	1·28	2·86	5·10	8·00	11·5	15·6	20·4	25·8	31·8	38 5	45·8
" 12°	0·35	1·40	3·15	5·60	8·75	12·6	17·3	22·4	28·4	35·0	42·8	50·4
" 14°	0·39	1·56	3·50	6·22	9·70	14·0	19·0	25·0	31·5	39·0	47.0	56·0
" 16°	0·44	1·75	3·94	7·00	11·0	15·8	21·5	28·0	35·5	43·7	53·0	63·0
" 18°	0·50	2·00	4·50	8·00	12·5	18·0	24·5	32·0	40·5	50·0	60·5	72·0
" 20°	0·58	2·32	5·25	9·30	14·6	21·0	28·5	37·3	47·2	58·3	70·5	84·0
" 22°	0·70	2·80	6·30	11·2	17·5	25·2	34·3	44·8	56·7	70·0	84·7	100·0
" 24°	0·88	3·50	7·86	14·0	21·0	31·5	42·8	56·0	71·0	87·5	106·0	126·0

TABLE SHOWING REFRIGERATING EFFECT OF ONE CUBIC FOOT OF AMMONIA GAS AT DIFFERENT CONDENSER AND SUCTION (BACK) PRESSURE IN B.T. UNITS.

(From the "Compend of Mechanical Refrigeration.")

Temperature of Gas in Degrees F.	Corresponding Suction Pressure. lbs. per sq. in.	Temperature of the Liquid in Degrees F								
		65	70	75	80	85	90	95	100	105
		Corresponding Condenser Pressure (Gauge) lbs. per square inch.								
		103	115	127	139	153	168	184	200	218
	G. Pres									
—27	1	27·30	27·01	26·73	26·44	26·16	25·87	25·59	25·30	25·02
—20	4	33·74	33·40	33·04	32·70	32·34	31·99	31·64	31·30	30·94
—15	6	36·86	36·48	36·10	35·72	35·34	34·96	34·58	34·20	33·82
—10	9	43·28	41·84	41·41	40·97	40·54	40·10	39·67	39·23	38·80
— 5	13	48·31	47·81	47·32	46·82	46·33	45·83	45·34	44·84	44·35
0	16	54 88	54·32	53·76	53·20	52·64	52·08	51·52	50·96	50·40
5	20	61·50	60·87	60·25	59 62	59·00	58·37	57·75	57·12	56·50
10	24	68·66	67·97	67·27	66·58	65·88	65·19	64·49	63·80	63·10
15	28	75·88	75·12	74·35	73·59	72·82	72·06	71·29	70·53	69·76
20	33	85·15	84·30	83·44	82·59	81·73	80·88	80·02	79·17	78·31
25	39	95·50	94·54	93·59	92·63	91·68	90·72	89·97	88·81	87·86
30	45	106·21	105·15	104·09	103·03	101·97	100·91	99·85	98·79	97·73
35	51	115 69	114·54	123·39	112·24	111·09	109·94	108·79	107·64	106·49

COEFFICIENT OF EXPANSION FROM 32° TO 210° F.

Glass	0·000,861,30	Pine wood (lengthwise) ..	0·000,3
Platinum	0·000,884,20	Oak wood	0·000,7
Steel, soft	0·001,078,80	Granite	0·000,8
Iron, cast	0·001,125,00	Limestone	0·000,8
Iron, wrought	0·001,220,40	Antimony	0·001,1
Steel, hardened	0·001,239,50	Gold	0·001,4
Copper	0·001,718,20	Ebonite	0·001,7
Bronze	0·001,816,70	Nickel	0·001,8
Brass	0·001,878,20	Silver	0·001,9
Tin	0·002,173,00	Aluminium	0·002,3
Lead	0·002,857,50	Pine wood (crosswise)	0·005,8
Zinc	0·002,941,70	Mercury (in glass tube) ..	0·016,2

PROPERTIES OF SATURATED AMMONIA GAS.

Temperature of Ebullition.	Absolute Pressure per square inch.	Heat of Liquid reckoned from 32° Fahr.	Latent Heat of Evaporation.	Increase of Volume during Evaporation.	Density of Vapour or Weight of one cubic foot.
Deg. Fahr.	lbs.	B.T.U.	B.T.U.	Cubic feet.	lb.
—40	10·22	—67·15	603·45	25·05	·040
—31	13·23	—59·33	599·25	19·70	·051
—22	16·95	—51·35	594·86	15·64	·064
—13	21·51	—43·20	590·29	12·53	·080
— 4	27·04	—34·80	585·51	10·14	·098
5	33·67	—26·42	580·56	8·27	·121
14	41·58	—17·78	575·40	6·79	·147
23	50·91	— 8·97	570·06	5·64	·177
32	61·85	0·00	564·53	4·71	·211
41	74·55	9·14	558·81	3·96	·251
50	89·21	18·43	552·92	3·36	·296
59	105·99	27·90	546·81	2·87	·346
68	125·08	37·53	540·51	2·47	·402
77	146·64	47·32	534·04	2·13	·465
86	170·83	57·28	527·36	1·86	·534
95	197·83	67·40	520·50	1·62	·610
104	227·76	77·69	513·46	1·43	·691

PROPERTIES OF SATURATED CARBONIC ACID GAS.

Temperature of Ebullition.	Absolute Pressure per square inch.	Heat of Liquid reckoned from 32° Fahr.	Latent Heat of Evaporation.	Increase of Volume during Evaporation.	Density of Vapour or Weight of one cubic foot.
Deg. Fahr.	lbs.	B.T.U.	B.T.U.	Cubic feet.	lbs.
—22	210	—37·80	136·15	·4138	2·321
—13	249	—32·51	131·65	·3459	2·759
— 4	292	—26·91	126·79	·2901	3·265
5	342	—20·92	121·50	·2435	3·853
14	396	—14·49	115·70	·2042	4·586
23	457	— 7·56	109·37	·1711	5·331
32	525	0·00	102·35	·1426	6·265
41	599	8·32	94·52	·1177	7·374
50	680	17·60	85·64	·0960	8·708
59	768	28·22	75·37	·0768	10·356
68	864	40·86	62·98	·0577	12·480
77	968	57·06	46·89	·0391	15·475
86	1080	84·44	19·28	·0147	21·519

MAXIMUM STRENGTH OF LIQUOR AMMONIA AT DIFFERENT TEMPERATURES.

Degrees Fahrenheit.	Pound of NH_3 to 1 lb. of Water.	Degrees Fahrenheit.	Pound of NH_3 to 1 lb. of Water.
32	0·875	68	0·526
35·6	0·833	71·6	0·499
39·2	0·792	75·2	0·474
42·8	0·751	78·8	0·449
46·4	0·713	82·4	0·426
50	0·679	86	0·403
53·6	0·645	89·6	0·382
57·2	0·612	93·2	0·362
60·8	0·582	96·8	0·343
64·4	0·554	100·4	0·324

WOOD'S TABLE OF SATURATED AMMONIA.

Recalculated by George Davidson, M.E.

Temperature. Degrees F. t.	Temperature. Absolute T.	Pressure Absolute. Pounds per sq. foot P.	Pressure Absolute. Pounds per sq. inch p.	Gauge Pressure, pound per sq. inch.	Heat of Vaporisation, thermal units h.	Volume of Vapour per pound, cubic feet v.	Volume of Liquid per pound, cubic feet v_1.	Weight of Vapour in pounds per cubic foot w.	Weight of Liquid in pounds per cubic foot w_1.	Temperature. Degrees F. t
—40	420·66	1539 90	10·69	—4·01	579·67	24·388	·02348	·0410	42·589	—40
39	1	1584·43	11·00	— 3·70	579 07	23·785	·02351	·0421	42·535	39
38	2	1630 08	11 32	—3·38	578·42	23·102	·02354	·0433	42·483	38
37	3	1676·71	11·64	— 3·06	577·88	22·488	·02357	·0444	42·427	37
36	4	1724·51	11·98	—2·72	577·27	21·895	·02359	·0457	42·391	36
—35	425·66	1773·43	12·31	—2·39	576·68	21·321	·02362	·0469	42·337	—35
34	6	1823·50	12·66	—2·04	576·08	20·768	·02364	·0482	42·301	34
33	7	1874·73	13·02	—1·68	575·48	20·231	·02366	·0495	42·265	33
32	8	1927·17	13·38	—1·32	574·89	19·708	·02368	·0507	42·218	32
31	9	1980·78	13·75	—0·95	574·39	19·204	·02371	·0521	42·176	31
—30	430·66	2035·69	14·13	—0·57	573·69	18·693	·02374	·0535	42·123	—30
29	1	2091·83	14·53	—0·17	573·08	18·225	·02378	·0549	42·052	29
28	2	2149·23	14·92	+0·22	572·48	17·759	·02381	·0563	42·000	28
27	3	2207·94	15·33	+0·63	571·89	17·307	·02384	·0577	41·946	27
26	4	2267·97	15·75	+1 05	571·28	16·869	·02387	·0593	41·893	26
—25	435·66	2329·34	16·17	+1·47	570·68	16·446	·02389	·0608	41·858	—25
24	6	2392·09	16·61	1·91	570·08	16·034	·02392	·0624	41·806	24
23	7	2456·23	17·05	2·35	569·48	15·633	·02395	·0640	41·754	23
22	8	2520·45	17·50	2·8	568·88	15·252	·02398	·0656	41·701	22
21	9	2588·77	17·97	3·27	568·27	14·875	·02401	·0672	41·649	21
—20	440·66	2657·23	18·45	+3·75	567·67	14·507	·02403	·0680	41·615	—20
19	1	2727·17	18·94	4·24	567·06	14·153	·02406	·0706	41·563	19
18	2	2798·62	19·43	4·73	566·48	13·807	·02409	·0725	41·511	18
17	3	2871 61	19·94	5·24	565·85	13·475	·02411	·0742	41·480	17
16	4	2946·17	20·46	5·76	565·25	13·150	·02414	·0760	41·425	16
—15	445·66	3022·31	20·99	+6·29	564·64	12·834	·02417	·0779	41·374	—15
14	6	3100·07	21·53	6·83	564·04	12·527	·02420	·0798	41·322	14
13	7	3179·45	22·08	7·38	563·43	12·230	·02423	·0818	41·271	13
12	8	3260·52	22·64	7·94	562·82	11·939	·02425	·0838	41·237	12
11	9	3343·29	23·22	8·52	562·21	11·659	·02428	·0858	41·186	11
—10	450·66	3427·75	23·80	+9·10	561·61	11·385	·02431	·0878	41·135	—10
9	1	3513·97	24·40	9·70	560·99	11·117	·02434	·0899	41·084	9
8	2	3601·97	25·01	10·31	560·39	10·860	·02437	·0921	41·034	8
7	3	3691·75	25·64	10·94	559·78	10·604	·02439	·0943	41·000	7
6	4	3783·37	26·27	11·57	559·17	10·362	·02442	·0965	40 950	6
— 5	455·66	3876·85	26·92	+12·22	558·56	10·125	·02445	·0988	40 900	— 5
4	6	3972·62	27·59	12·89	557·94	9·894	·02448	·1011	40·845	4
3	7	4069·48	28·26	13·56	557 33	9·669	·02451	·1034	40·799	3
2	8	4168·70	28·95	14·25	556·73	9·449	·02454	·1058	40·749	2
1	9	4269·90	29·65	14·95	556·11	9·234	·02457	·1083	40·700	1
+ 0	460·66	4372·10	30·37	+15·67	555·50	9·028	·02461	·1107	40·650	+ 0
1	1	4478·82	31·10	16·40	554·88	8·825	·02463	·1133	40·601	1
2	2	4585·60	31·84	17·14	554·27	8·630	·02466	·1159	40·551	2
3	3	4694·96	32 60	17·90	553·65	8·436	·02469	·1186	40·502	3
4	4	4806·46	33·38	18·68	553·04	8 250	·02472	1212	40·453	4

WOOD'S TABLE OF SATURATED AMMONIA (*continued*).

Temperature.		Pressure Absolute.		Gauge Pressure, pound per sq. inch.	Heat of Vaporisation, thermal units. h.	Volume of Vapour per pound cubic feet. v.	Volume of Liquid per pound cubic feet. v_1.	Weight of Vapour in pounds per cubic foot. w.	Weight of Liquid in pounds per cubic foot. w_1.	Temperature. Degrees F.
Degrees F. t.	Absolute. T.	Pounds per sq. foot. P.	Pounds per sq. inch. p.							
+ 5	465·66	4920·11	34·16	+19·46	552·43	8·070	·02475	·1240	40·404	+ 5
6	6	5035·95	34·97	20·27	551·81	7·892	·02478	·1267	40·355	6
7	7	5153·99	35·79	21·09	551·19	7·717	·02480	·1296	40·322	7
8	8	5274·28	36·63	21·93	550·58	7·553	·02483	·1324	40 274	8
9	9	5396·88	37·48	22·78	549·96	7·388	·02486	·1353	40·225	9
+10	470·66	5521·71	38·34	+23·64	549·35	7·229	·02490	·1383	40·160	+10
11	1	5649·48	39·23	24·53	548·73	7·075	·02493	·1413	40·112	11
12	2	5778·50	40·18	25·43	548·11	6·924	·02496	·1444	40 064	12
13	3	5910·52	41·04	26·34	547·49	6·786	·02499	·1474	40·016	13
14	4	6044·96	41·98	27·28	546·88	6·632	·02502	·1507	39 968	14
+15	475·66	6182·00	42·94	+28·24	546·26	6·491	·02505	·1541	39·920	+15
16	6	6321·24	43·90	29·20	545 63	6·355	·02508	·1573	39·872	16
17	7	6463·24	44·88	30·18	545·01	6·222	·02511	·1607	39·872	17
18	8	6607·77	45·89	31·19	544·39	6·098	·02514	·1641	39·777	18
19	9	6754·90	46·91	32·21	543·74	5·966	·02517	·1676	39·729	19
+20	480·66	6904·68	47·95	+33·25	543·15	5·848	·02520	·1711	39·682	+20
21	1	7057·15	49·01	34·31	542·53	5·722	·02523	·1748	39·635	21
22	2	7211·38	50·09	35·39	541·90	5·605	·02527	·1784	39·572	22
23	3	7370·27	51·18	36·48	541·28	5·488	·02529	·1822	39·541	23
24	4	7530·96	52·30	37·60	540·66	5·378	·02533	·1860	39·479	24
+25	485·66	7694·52	53·43	+38·73	540·03	5·270	·02536	·1897	39·432	+25
26	6	7860·89	54·59	39·89	539·41	5·163	·02539	·1937	39·386	26
27	7	8030·16	55·76	41·06	538·78	5·058	·02542	·1977	39·339	27
28	8	8202·38	56·96	42·26	538·16	4·960	·02545	·2016	39·292	28
29	9	8377·56	58·17	43·47	537·53	4·858	·02548	·2059	39·246	29
+30	490·66	8555·74	59·42	+44·72	536·91	4·763	·02551	·2099	39·200	+30
31	1	8736·96	60·67	45·97	536·28	4·668	·02554	·2142	39·115	31
32	2	8921·26	61·95	47·25	535·66	4·577	·02557	·2185	39·108	32
33	3	9108·71	63·25	48·55	535·03	4·486	·02561	·2229	39·047	33
34	4	9299·32	64·58	49·88	534·40	4·400	·02564	·2273	39·001	34
+35	495·66	9493·07	65·92	+51·22	533·78	4·314	·02568	·2318	38·940	+35
36	6	9690·04	67·29	52·59	533·13	4·234	·02571	·2362	38·894	36
37	7	9890·75	68·68	53·98	532·52	4·157	·02574	·2415	38·850	37
38	8	10093·91	70·09	55·39	531·89	4·068	·02578	·2458	38·789	38
39	9	10300·88	71·53	56·83	531·26	3·989	·02582	·2507	38·729	39
+40	500·66	10511·16	72·99	+58·29	530·63	3·915	·02585	·2554	38·684	+40
41	1	10724·95	74·48	59·78	529·99	3·839	·02588	·2605	38·639	41
42	2	10942·18	75·99	61·29	529·36	3·766	·02591	·2655	38·595	42
43	3	11162·93	77·52	62·82	528·73	3·695	·02594	·2706	38·550	43
44	4	11387·21	79·08	64·38	528·10	3·627	·02597	·2757	38·499	44
+45	505·66	11615·12	80·66	+65·96	527·47	3·559	·02600	·2809	38·461	+45
46	6	11846·64	82·27	67·57	526·83	3·493	·02603	·2863	38·417	46
47	7	12081·80	83·90	69·20	526·20	3·428	·02606	·2917	38·373	47
48	8	12320·71	85·56	70·86	525·57	3·362	·02609	·2974	38·328	48
49	9	12563·36	87·25	72·55	524·93	3·303	·02612	·3027	38·284	49
+50	510·66	12809·91	88·96	+74·26	524·30	3·242	·02616	·3084	38·226	+50
51	1	13060·21	90·70	76·00	523·66	3·182	·02620	·3143	38·167	51
52	2	13314·43	92·46	77·76	523·03	3·124	·02623	·3201	38·124	52
53	3	13572·52	94·25	79·55	522·39	3·069	·02626	·3258	38·080	53
54	4	13834·64	96·07	81·37	521·76	3·012	·02629	·3320	38·037	54

WOOD'S TABLE OF SATURATED AMMONIA (*continued*).

Temperature.		Pressure, Absolute.		Gauge Pressure, pound per sq. inch.	Heat of Vaporisation, thermal units. h.	Volume of Vapour per pound cubic feet. v.	Volume of Liquid per pound cubic feet. v_1.	Weight of Vapour in pounds per cubic feet. w.	Weight of Liquid in pounds per cubic feet. w_1.	Temperature. Degrees F.
Degrees F. t.	Absolute. T.	Pounds per sq. feet. p.	Pounds per sq. inch. p.							
+ 55	515·66	14100·74	97·92	+ 83·23	521·12	2·958	·02632	·3380	37·994	+ 55
56	6	14370·92	99·80	85·10	520·48	2·905	·02636	·3442	37·936	56
57	7	14645·18	101·70	87·00	519·84	2·853	·02639	·3505	37·893	57
58	8	14923·96	103·64	88·94	519·20	2·802	·02643	·3568	37·835	58
59	9	15206·28	105·60	90·90	518·57	2·753	·02646	·3632	37.793	59
+ 60	520·66	15493·09	107·59	+ 92·89	517·93	2·705	·02650	·3697	37·736	+ 60
61	1	15784·23	109·61	94·91	517·29	2·658	·02654	·3762	37·678	61
62	2	16079·67	111·66	96·96	516·65	2·610	·02658	·3831	37·622	62
63	3	16379·51	113·75	99·05	516·01	2·565	·02661	·3898	37·579	63
64	4	16683·75	115·86	101·16	515·37	2·520	·02665	·3968	37·523	64
+ 65	525·66	16992·50	118·03	+103·33	514·73	2·476	·02668	·4039	37·481	+ 65
66	6	17305·70	120·18	105·48	514·09	2·433	·02671	·4110	37·439	66
67	7	17623·45	122·38	107·68	513·45	2·389	·02675	·4189	37·383	67
68	8	17945·89	124·62	109 92	512·81	2·351	·02678	·4254	37 341	68
69	9	18272·81	126·89	112·19	512·16	2·310	·02682	·4329	37·285	69
+ 70	530·66	18604·53	129·19	+114·49	511·52	2·272	·02686	·4401	37·230	+ 70
71	1	18941·00	131·54	116·84	510·87	2·233	·02689	·4479	37·188	71
72	2	19282·21	133·90	119·20	510·22	2·194	·02693	·4558	37·133	72
73	3	19628·32	136·31	121·61	509·58	2·158	·02697	·4645	37·079	73
74	4	19979·22	138·74	124·04	508·93	2·122	·02700	·4712	37·037	74
+ 75	535·66	20335·16	141·22	+126·52	508·29	2·087	·02703	·4791	36·995	+ 75
76	6	20696·00	143·72	129·02	507·64	2·052	·02706	·4873	36·954	76
77	7	21061·85	146·26	131·56	506·99	2·017	·02710	·4957	36·900	77
78	8	21432·82	148·84	134·14	506·34	1·985	·02714	·5012	36·845	78
79	9	21806·85	151·45	136·75	505·69	1·952	·02717	·5123	36·805	79
+ 80	540·66	22190·15	154·10	+139·40	505·05	1·921	·02721	·5233	36·751	+ 80
81	1	22576·51	156·78	142·08	504·40	1·889	·02725	·5294	36·696	81
82	2	22968·88	159·50	144·80	503·75	1·850	·02728	·5382	36·557	82
83	3	23365·38	162·26	147·56	503·10	1·827	·02732	·5478	36·603	83
84	4	23767·81	165·05	150·35	502·45	1·799	·02736	·5558	36·549	84
+ 85	545·66	24175·61	167·88	+153·18	501·81	1·770	·02739	·5649	36·509	+ 85
86	6	24538·92	170 75	156·05	501·15	1·741	·02743	·5744	36·456	86
87	7	25007·80	173·66	158·96	500·50	1·714	·02747	·5834	36·407	87
88	8	25432·16	176·61	161.91	499·85	1·687	·02751	·5927	36·350	88
89	9	25862·14	179·59	164·89	499·20	1·660	·02754	·6024	36·311	89
+ 90	550·66	26297·88	182·62	+167·92	498·55	1·634	·02758	·6120	36·258	+ 90
91	1	26739·88	185·69	170·99	497·89	1·608	·02761	·6219	36·219	91
92	2	27186·56	188·79	174·09	497·24	1·583	·02765	·6317	36·166	92
93	3	27639·48	191·94	177·24	496·59	1·558	·02769	·6418	36·114	93
94	4	28098·26	195·13	180·43	495·94	1·534	·02772	·6518	36·075	94
+ 95	555·66	28563·00	198·35	+183·65	495·29	1·510	·02776	·6623	36·023	+ 95
96	6	29033·86	201·62	186·92	494·63	1·486	·02780	·6729	35·971	96
97	7	29510·69	204·94	190·24	493·97	1·463	·02784	·6835	35·919	97
98	8	29993·52	208·29	193·59	493·32	1·442	·02787	·6934	35·881	98
99	9	30482·52	211·68	196·98	492·66	1·419	·02791	·7047	35·829	99
+100	560·66	30977·78	215·12	+200·42	492·01	1·398	·02795	·7153	35·778	+100

COMPARISON OF THERMOMETER SCALES.

R.	C.	F.	R.	C.	F.
+80	+100	+212	+23	+28·75	+83·75
79	98·75	209·75	22	27·50	81·50
78	97·50	207·50	21	26·25	79·25
77	96·25	205·25	20	25	77
76	95	203	19	23·75	74·75
75	93·75	200·75	18	22·50	72·50
74	92·50	198·50	17	21·25	70·25
73	91·25	196·25	16	20	68
72	90	194	15	18·75	65·75
71	88·75	191·75	14	17·50	63·50
70	87·50	189·50	13	16·25	61·25
69	86·25	187·25	12	15	59
68	85	185	11	13·75	56·75
67	83·75	182·75	10	12·50	54·50
66	82·50	180·50	9	11·25	52·25
65	81·25	178·25	8	10	50
64	80	176	7	8·75	47·75
63	78·75	173·75	6	7·50	45·50
62	77·50	171·50	5	6·25	43·25
61	76·25	169·25	4	5	41
60	75	167	3	3·75	38·75
59	73·75	164·75	2	2·50	36·50
58	72·50	162·50	1	1·25	34·25
57	71·25	160·25	0	0	32
56	70	158	—1	—1·25	29·75
55	68·75	155·75	2	2·50	27·50
54	67·50	153·50	3	3·75	25·25
53	66·25	151·25	4	5	23
52	65	149	5	6·25	20·75
51	63·75	146·75	6	7·50	18·50
50	62·50	144·50	7	8·75	16·25
49	61·25	142·25	8	10	14
48	60	140	9	11·25	11·75
47	58·75	137·75	10	12·50	9·50
46	57·50	135·50	11	13·75	7·25
45	56·25	133·25	12	15	5
44	55	131	13	16·25	2·75
43	53·75	128·75	14	17·50	0·50
42	52·50	126·50	15	18·75	—1·75
41	51·25	124·25	16	20	4
40	50	122	17	21·25	6·25
39	48·75	119·75	18	22·50	8·50
38	47·50	117·50	19	23·75	10·75
37	46·25	115·25	20	25	13
36	45	113	21	26·25	15·25
35	43·75	110·75	22	27·50	17·50
34	42·50	108·50	23	28·75	19·75
33	41·25	106·25	24	30	22
32	40	104	25	31·25	24·25
31	38·75	101·75	26	32·50	26·50
30	37·50	99·50	27	33·75	28·75
29	36·25	97·25	28	35	31
28	35	95	29	36·25	33·25
27	33·75	92·75	30	37·50	35·50
26	32·50	90·50	31	38·75	37·75
25	31·25	88·25	32	40	40
24	30	86			

TEMPERATURES—FAHRENHEIT AND CENTIGRADE.

°F.	°C.	°F.	°C.	°F.	°C.	°F.	°C.	°F.	°C.	°F.	°C.
330	165·6	267	130·6	206	96·7	143	61·7	80	26·7	19	— 7·2
329	165·	266	130·	205	96·1	142	61·1	79	26·1	18	— 7·8
328	164·4	265	129·4	204	95·6	141	60·6	78	25·6	17	— 8·3
327	163·9	264	128·9	203	95·	140	60·	77	25·	16	— 8·9
326	163·3	263	128·3	202	94·4	139	59 4	76	24·4	15	— 9·4
325	162·8	262	127·8	201	93·9	138	58·9	75	23·9	14	—10
324	162·2	261	127·2	200	93·3	137	58·3	74	23·3	13	—10·6
323	161·7	260	126·7	199	92·8	136	57·8	73	22·8	12	—11·1
322	161·1	259	126·1	198	92·2	135	57·2	72	22·2	11	—11·7
321	160·6	258	125·6	197	91·7	134	56·7	71	21·7	10	—12·2
320	160·	257	125·	196	91·1	133	56·1	70	21·1	9	—12·8
319	159·4	256	124·4	195	90·6	132	55·6	69	20·6	8	— 13·3
318	158·9	255	123·9	194	90·	131	55·	68	20·	7	—13·9
317	158·3	254	123·3	193	89·4	130	54·4	67	19·4	6	—14·4
316	157·8	253	122·8	192	88·9	129	53·9	66	18·9	5	—15·
315	157·2	262	122·2	191	88·3	128	53·3	65	18·3	4	—15·6
314	156·7	251	121·7	190	87·8	127	52·8	64	17·8	3	—16·1
313	156·1	250	121·1	189	87·2	126	52·2	63	17·2	2	—16·7
312	155·6	249	120·6	188	86·7	125	51·7	62	16·7	1	— 17·2
311	155·	248	120·	187	86·1	124	51·1	61	16·1	0	—17·8
310	154·4	247	119·4	186	85·6	123	50·6	60	15·6	— 1	—18·3
309	153·9	246	118·9	185	85·	122	50·	59	15	— 2	—18·9
308	153·3	245	118·3	184	84·4	121	49·4	58	14·4	— 3	— 19·4
307	152·8	244	117·8	183	83·9	120	48·9	57	13·9	— 4	—20·
306	152·2	243	117·2	182	83·3	119	48·3	56	13·3	— 5	— 20·6
305	151·7	242	116·7	181	82·8	118	47·8	55	12·8	— 6	— 21·1
304	151·1	241	116·1	180	62·2	117	47·2	54	12·2	— 7	—21·7
303	150·6	240	115·6	179	81·7	116	46·7	53	11·7	— 8	—22·2
302	150·	239	115·	178	81·1	115	46·1	52	11·1	— 9	—22·8
301	149·4	238	114·4	177	80·6	114	45·6	51	10·6	—10	—23·3
300	148·9	237	113·9	176	80·	113	45·	50	10·	—11	—23·9
299	148·3	236	113·3	175	79 4	112	44·4	49	9·4	—12	—24·4
298	147·8	235	112·8	174	78·9	111	43·9	48	8·9	—13	—25·
297	147·2	234	112·2	173	78·3	110	43·3	47	8·3	—14	—25·6
296	146·7	233	111·7	172	77·8	109	42·8	46	7·8	—15	—26·1
295	146·1	232	111·1	171	77·2	108	42·2	45	7·2	—16	—26·7
294	145·6	231	110·6	170	76·7	107	41·7	44	6·7	—17	—27·2
293	145·	230	110·	169	76·1	106	41·1	43	6·1	—18	—27·8
292	144·4	229	109·4	168	75·6	105	40·6	42	5·6	—19	—28·3
291	143·9	228	108·9	167	75·	104	40·	41	5·	—20	—28·9
290	143·3	227	108·3	166	74·4	103	39·4	40	4·4	—21	—29·4
289	142·8	226	107·8	165	73·9	102	38·9	39	3·9	—22	—30·
288	142·2	225	107·2	164	73·3	101	38·3	38	3·3	—23	—30·6
287	141·7	224	106·7	163	72·8	100	37·8	37	2·8	—24	—31·1
286	141·1	223	106·1	162	72·2	99	37·2	36	2·2	—25	—31·7
285	140·6	222	105·6	161	71·7	98	36·7	35	1·7	—26	—32·2
284	140·	221	105·	160	71·1	97	36·1	34	1·1	—27	—32·8
283	139·4	220	104·4	159	70·6	96	35·6	33	0·6	—28	—33·3
282	138·9	219	103·9	158	70·	95	35·			—29	—33·9
281	138·3	218	103·3	157	69·4	94	34·4	Water freezes.		—30	—34·4
280	137 8	217	102·8	156	68·9	93	33·9	32	0·	—31	—35·
279	137·2	216	102·2	155	68·3	92	33·3	31	— 0·6	—32	—35·6
278	136·7	215	101·7	154	67·8	91	32·8	30	— 1·1	—33	—36·1
277	136·1	214	101·1	153	67·2	90	32·2	29	— 1·7	—34	—36·7
276	135·6	213	100·6	152	66·7	89	31·7	28	— 2·2	—35	—37·2
275	135·			151	66·1	88	31·1	27	— 2·8	—36	—37·8
274	134·4	Water boils.		150	65·6	87	30·6	26	— 3·3	—37	—38·3
273	133·9	212	100.	149	65·	86	30·	25	— 3·9	—38	—38·9
272	133 3	211	99·4	148	64·4	85	29·4	24	— 4·4	—39	—39·4
271	132·8	210	98·9	147	63·9	84	28·9	23	— 5·		
270	132·2	209	98·3	146	63·3	83	28·3	22	— 5·6	Mercury freezes.	
269	131·7	208	97·8	145	62·8	82	27·8	21	— 6·1	—40	—40
268	131·1	207	97·2	144	62·2	81	27·2	20	— 6·7		

SOLUBILITY OF AMMONIA IN 1 LB. OF WATER AT DIFFERENT TEMPERATURES AND PRESSURES.

Absolute Pressure in lbs. per square inch.	32° F.	68° F.	104° F.	212° F.
	lbs.	lb.	lb.	lb.
14·07	0·899	0 518	0·336	0·074
15·44	0 937	0·535	0·340	0·078
16·41	0·980	0·556	0·365	0·083
17·37	1·029	0·574	0·378	0·088
18·34	1·077	0·594	0·391	0·092
19·30	1·126	0·613	0·404	0 096
20·27	1·177	0·632	0·414	0·101
21·23	1·236	0·651	0·425	0·106
22·19	1·283	0·669	0·434	0·110
23·16	1·336	0·685	0·445	0·115
24·13	1·388	0·704	0·454	0·120
25·09	1·442	0·722	0·463	0·125
26·06	1·496	0·741	0·472	0·130
27·02	1·549	0 761	0·479	0·135
27·99	1·603	0·780	0·486	
28·95	1·656	0·801	0·493	
30·88	1·758	0·842	0·511	
32·81	1·861	0·881	0 530	
34·74	1·966	0·919	0·547	
36 67	2·070	0·955	0·565	
38 60		0·992	0·579	
40·53			0·591	

STRENGTH OF AMMONIA LIQUOR.

Percentage of Ammonia by Weight.	Specific Gravity.	Degrees Beaumé Water 10.	Degrees Beaumé Water 0.
0	1·000	10	0
1	0·993	11	1
2	0·986	12	2
4	0·979	13	3
6	0 972	14	4
8	0·966	15	5
10	0·960	16	6
12	0·953	17·1	7
14	0·945	18·3	8 2
16	0·938	19·5	9·2
18	0·931	20·7	10·3
20	0·925	21·7	11·2
22	0·919	22·8	12·3
24	0·913	23·9	13·2
26	0·907	24·8	14·3
28	0·902	25·7	15·2
30	0·897	26·6	16·2
32	0·892	27·5	17·3
34	0·888	28·4	18·2
36	0·884	29·3	19·1
38	0·880	30·2	20·0

SOLUBILITY OF AMMONIA IN WATER AT DIFFERENT TEMPERATURES (ROSCOE).

Degrees Celsius.	Degrees Fahrenheit.	Pounds of NH_3 to 1 lb. Water.	Degrees Celsius.	Degrees Fahrenheit.	Pounds of NH_3 to 1 lb. Water.
0	32·	0·875	28	82·4	0·426
2	35·6	0·833	30	86·	0·403
4	39·2	0·792	32	89·6	0·382
6	42·8	0·751	34	93·2	0·362
8	46·4	0·713	36	96·8	0·343
10	50·	0·679	38	100·4	0·324
12	53·6	0·645	40	104·0	0·307
14	57·2	0·612	42	107·6	0·290
96	60·8	0·582	44	111·2	0·275
18	64·4	0·554	46	114·8	0·259
20	68·	0·526	48	118·4	0·244
22	71·6	0·499	50	122·	0·229
24	75·2	0·474	52	125·6	0·214
26	78·8	0·449	54	129·2	0·200
			56	132·8	0·186

COMPARATIVE ABSORBING OR RADIATING AND REFLECTING PROPERTIES OF SOLIDS.

Substance.	Absorbing or Radiating Power.	Reflecting Power.
	Proportion, per cent.	Proportion, per cent.
Brass, bright polished	7	93
Brass, dead polished	11	89
Copper	7	93
Glass	90	10
Gold	5	95
Ice	85	15
Iron, cast, polished	25	75
Iron, wrought, polished	23	77
Marble	93 to 98	7 to 2
Mercury	23	77
Platinum, polished	24	76
Platinum, sheet	17	83
Silver leaf on glass	27	73
Silver, polished	3	97
Steel, polished	17	83
Tin	15	85
Water	100	0
Writing paper	98	2
Zinc, polished	19	81

TABLE SHOWING LATENT HEAT OF FUSION.

	Thermal Units.		Thermal Units.
Ice	142·5	Tin	25·5
Nitrate of ammonia	113·2	Cadmium	24·5
Nitrate of soda	104·1	Bismuth	22·7
Phosphate of potash	85·1	Sulphur	16·8
Nitrate of potash	78·4	Lead	9·5
Chloride of calcium	64·3	Phosphorus	9·0
Zinc	50·6	D'Arcet's alloy	8·1
Platinum	48·8	Mercury	5·1
Silver	37·8		

MEAN PRESSURE OF COMPRESSOR.

Condenser Pressure.		108	115	127	139	153	168	184	200	218
Condenser Temperature		65°	70°	75°	80°	85°	90°	95°	100°	105°
Refrigerator Pressure.	Refrigerator Temperature.									
4	—20°	41·46	43·91	46·34	48·77	51·23	53·68	56·11	58·54	60·99
6	—15°	42·72	45·38	47·90	50·74	53·40	56·08	58·86	61·40	64·08
9	—10°	44·40	47·38	50·38	53·29	56·25	59·20	62·16	65·14	68·09
13	— 5°	45·86	49·15	52·42	55·70	58·97	62·25	65·53	68·81	72·08
16	0°	46·94	50·56	54·16	57·78	61·40	65·00	68·62	72·22	75·84
20	5°	47·74	51·73	55·70	59·68	63·67	67·66	71·62	75·61	79·61
24	10°	48·04	52·40	56·77	61·13	65·51	69·86	74·24	78·59	82·97
28	15°	47·88	52·67	57·44	62·23	67·02	71·81	76·60	81·39	86·18
33	20°	47·06	52·30	57·53	62·75	67·98	73·23	78·46	83·68	88·91
39	25°	46·06	51·34	57·05	62·75	68·46	74·17	79·88	85·58	91·29
45	30°	43·16	49·71	55·92	62·14	68·35	74·56	80·77	86·98	93·19
51	35°	40·52	47·26	54·02	60·76	67·52	74·26	81·02	87·78	94·52

TABLE GIVING NUMBER OF CUBIC FEET OF GAS THAT MUST BE PUMPED PER MINUTE AT DIFFERENT CONDENSER AND SUCTION PRESSURES, TO PRODUCE ONE TON OF REFRIGERATION IN TWENTY-FOUR HOURS.

(From the "Compend of Mechanical Refrigeration.")

Temperature of Gas in Degrees F.	Corresponding Suction Pressure. lbs. per sq. in.	Temperature of the Gas in Degrees F								
		65	70	75	80	85	90	95	100	105
		Corresponding Condenser Pressure (Gauge) lbs. per square inch.								
		103	115	127	139	153	168	184	200	218
	U. Pres									
—27	1	7·22	7·3	7·37	7·46	7·54	7·62	7·70	7·79	7·88
— 20	4	5·84	5·9	5·96	6·03	6·09	6·16	6·23	6·30	6·43
—15	6	5·35	5·4	5·46	5·52	5·58	5·64	5·70	5·77	5·83
—10	9	4·66	4·73	4·76	4·81	4·86	4·91	4·97	5 06	5·08
— 5	13	4·09	4·12	4·17	4.21	4·25	4 30	4·35	4·40	4·44
0	16	3·59	3·63	3·66	3·70	3·74	3·78	3·83	3·87	3·91
5	20	3·20	3·24	3·27	3·30	3·34	3·38	3·41	3·45	3·49
10	24	2 87	2·9	2·93	2·96	2·99	3·02	3·06	3·09	3·12
15	28	2·59	2·61	2·65	2·68	2·71	2·73	2·76	2·80	2·82
20	33	2·31	2·34	2·36	2 38	2·41	2·44	2·46	2·49	2·51
25	39	2·06	2·08	2·10	2·12	2·15	2·17	2·20	2·23	2·24
30	45	1·85	1·87	1·89	1·91	1·93	1·95	1·97	2·00	2 01
35	51	1·70	1·72	1·74	1·76	1·77	1·79	1·81	1·82	1·85

DIMENSIONS OF ICE-MAKING TANKS.

Table compiled by Skinkle, giving sizes of some freezing tanks, piping, and moulds in actual operation.

(From the "Compend of Mechanical Refrigeration.")

Tons, Ice-making Capacity.	No. of Tanks.	Sizes of Tanks. Length of Tank. Feet & Inches.	Width of Tank. Feet & Inches.	Depth of Tank in inches.	Thickness of Plates in inches.	No. of Coils.	Size of Pipe in inches.	No. of Pipes High.	Length of Coils.	Total feet of Pipe in Tank.	Feet of Pipe per ton Ice-making Capacity.	Number of Ice Moulds in Tank.	Size of Moulds in inches.	Net Weight of Ice from each Mould.	Number of Moulds per ton Ice-making Capacity.	Number Hours for Freezing each Mould.	Remarks.
1 2	1	17—0	6—2	33	3-16	7	1	6	15—4	614	322	60	8×15×33	100 lbs.	30	36	
3	1	17—0	9—0	33	3-16	10	1	6	15 0	900	300	90	8×15×33	100 ..	30	36	
5	1	17—0	14—9	33	3-16	16	1	6	15—0	1,440	288	150	8×15×33	100 ..	30	36	
10	1	29—0	19 0	33	¼	25	1	8	17—0	3,400	340	192 24	11×22×33 11×11×33	200 .. 100 ..	20·4	48	
12½	1	37—6	19 - 0	33	¼	33	1	8	17 - 0	4,488	329	256 32	11×22×33 11×11×22	200 .. 100 ..	21·36	53·2	Special.
15	1	48 - 0	19—0	33	¼	37	1	8	17—0	5,032	335	288 36	11×22×33 11×11×33	200 .. 100 ..	20·4	48	
20	2	29—0	19—0	33	¼	25	1	8	17—0	3,400	340	192 24	11×22×33 11×11×33	200 .. 100 ..	20·4	48	*
30	2	48—0	19—0	33	¼	37	1	8	17—0	5,032	335	288 36	11×22×33 11×11×33	200 .. 100 ..	20·4	48	*
30	1	43 - 0	30 - 0	48	¼	35	1¼	8	28—0	7,840	261	480	11×22×45	300 ..	16	57·6	
60	2	56—0	20—0	48	¼	49	1¼	10	18—0	8,820	294	432	11×22×45	300 ..	14·4	51·8	
60	2	48 0	30—0	48	¼	35	1¼	8	28—0	7,840	261	480	11×22×45	300 ..	16	57·6	*

Average of 1in. pipe per ton, 327 feet. Average of 1¼in. pipe per ton, 272 feet.

* Twenty-ton tanks are duplicate 10-ton tanks }
Thirty-ton " " " 15 " " } Dimensions of one tank only are given in each instance.
Sixty-ton " " " 30 " " }

TABLE SHOWING PROPERTIES OF SOLUTION OF SALT.

(Chloride of Sodium.)

Percentage of Salt by Weight.	Pounds of Salt per gallon of Solution.	Degrees on Salometer at 60° F	Weight per gallon at 39° F = 4° C.	Specific Gravity at 39° F. = 4° C.	Specific Heat.	Freezing Point, Fahrenheit.	Freezing Point, Celsius.
1	0·084	4	8·40	1·007	0·992	30·5	— 0·8
2	0·169	8	8·46	1·015	0·984	29·3	— 1·5
2·5	0·212	10	8·50	1·019	0·980	28·6	— 1·9
3	0·256	12	8·53	1·023	0·976	27·8	— 2·3
3·5	0·300	14	8·56	1·026	0·972	27·1	— 2·7
4	0·344	16	8·59	1·030	0·968	26·6	— 3·0
5	0·433	20	8·65	1·037	0·960	25·2	— 3·8
6	0·523	24	8·72	1·045	0·946	23·9	— 4·5
7	0·617	28	8·78	1·058	0·932	22·5	— 5·3
8	0·708	32	8·85	1·061	0·919	21·2	— 6·0
9	0·802	36	8·91	1·068	0·905	19·9	— 6·7
10	0·897	40	8·97	1·076	0·892	18·7	— 7·4
12	1·092	48	9·10	1·091	0·874	16·0	— 8·9
15	1·389	60	9·26	1·115	0·855	12·2	—11·0
20	1·928	80	9·64	1·155	0·829	6·1	-14·4
24	2·376	96	9·90	1·187	0·795	1·2	—17·1
25	2·488	100	9·97	1·196	0·788	0·5	—17·8
26	2·610	104	10·04	1·204	0·771	— 1·1	—18·4

PROPERTIES OF SATURATED SULPHUR DIOXIDE GAS.

Temperature of Ebullition.	Absolute Pressure per square inch.	Heat of Liquid reckoned from 32° Fahr.	Latent Heat of Evaporation	Increase of Volume during Evaporation.	Density of Vapour or Weight of one cubic foot.
Deg. Fahr.	lbs.	B.T.U	B.T.U.	Cubic feet.	lb.
—22	5·56	—19·56	176·99	13·17	·076
—13	7·23	—16·30	174·95	10·27	·097
— 4	9·27	—13·05	172·89	8·12	·123
·5	11·76	— 9·79	170·82	6·50	·153
14	14·74	— 6·53	168·73	5·25	·190
23	18·31	— 3·27	166·63	4·29	·233
32	22·53	0·00	164·51	3·54	·282
41	27·42	3·27	162·38	2·93	·340
50	33·25	6·55	160·23	2·45	·407
59	39·93	9·83	158·07	2·07	·483
68	47·61	13·11	155·89	1·75	·570
77	56·39	16·39	153·70	1·49	·669
86	66·36	19·69	151·49	1·27	·780
95	77·64	22·98	149·26	1·09	·906
104	90·31	20·28	147·02	·91	1·046

TABLE SHOWING DIMENSIONS, ETC., OF ABSORPTION MACHINES.

Actual ice-making capacity in tons of ice	8	8	12	15	25	10
Number and size of steam boiler horse-power or dimensions ..	15	30	40″ × 20′	50	{2 42″ × 21½′	} 2 42″ × 10′
Pounds of coal used per hour ..	65	140	185	220	504	168-180
Number and size of generators ..	30″ × 10′	30″ × 16′	24″ × 18′	44″ × 14′	{2 30″ × 17½′	} 28″ × 15′
Size of coal in generator in square feet	24	48	91	96	400	80
Surface of discs, etc., in analyser in square feet	10	30	64	34	125	24
Cooling surface in exchanger in square feet	34	51	22½	68	65	25
Cooling surface of traps in absorber in square feet ..	180	360	191	470	1900	673
Cooling surface in condenser in square feet	345	690	220	1380	1220	544
Surface in expander or refrigerator in square feet ..	419	1200	726	2100	4000	1600
Cooling surface in rectifier in square feet	—	—	25	—	—	—
Cooling surface in heater ..	—	—	41	—	—	—
Temperature of water in degrees Fahrenheit	70	70	80	70	76	80-94
Temperature of brine in degrees Fahrenheit	10-20	10-20	10-12	10-20	7	10-14

TABLE SHOWING THE NUMBER OF BRANCHES EQUIVALENT TO A MAIN PIPE.

Size of Main Pipe.	Number of Branches.								
	2	3	4	5	6	7	8	9	10
Inches.									
1	·758	·644	·574	·525	·488	·459	·435	·415	·398
1¼	·965	·838	·747	·683	·635	·597	·556	·540	·518
1½	1·14	·967	·861	·788	·733	·689	·653	·623	·597
2	1·52	1·29	1·15	1·05	·977	·918	·870	·830	·796
2½	1·89	1·61	1·44	1·31	1·22	1·15	1·09	1·09	·995
3	2·27	1·92	1·72	1·58	1·47	1·38	1·31	1·25	1·19
3½	2·65	2·26	2·01	1·84	1·71	1·61	1·52	1·45	1·39
4	3·06	2·58	2·30	2·10	1·95	1·84	1·74	1·66	1·59
4½	3·41	2·90	2·58	2·36	2·20	2·07	1·96	1·87	1·79
5	3·79	3·22	2·87	2·63	2·44	2·30	2·18	2·08	1·99
6	4·55	3·87	3·45	3·15	2·93	2·75	2·61	2·49	2·39
7	5·30	4·51	4·02	3·68	3·42	3·21	3·05	2·91	2·79
8	6·06	5·16	4·59	4·20	3·91	3·67	3·48	3·32	3·18
9	6·82	5·80	5·17	4·73	4·40	4·13	3·92	3·74	3·58
10	7·58	6·44	5·74	5·25	4·88	4·59	4·35	4·15	3·98
12	9·08	7·73	6·89	6·30	5·86	5·51	5·22	4·98	4·78

DIMENSIONS OF CONDENSERS.

The following tables, compiled by Skinkle, give the dimensions of both submerged and atmospheric condensers of some plants in actual operation, and allow much more pipe for the atmospheric than for the submerged condenser:

ATMOSPHERIC CONDENSERS.

Ice-making Capacity in tons	Refrigerating Capacity in tons	Condenser Pans. Length of Pan in feet	Width of Pan in feet	Depth of Pan in inches	Thickness of Iron in inches	Number of Pipes High	Number of Pipes Wide	Size of Pipe in inches	Length of Coil over bends in feet	Total feet of Pipe in Condenser	Feet of Pipe per ton, Ice-making Capacity	Feet of Pipe per ton, Refrigerating Capacity
12½	25	21	10¾	8	3-16	40	5	1	17	8,680	294·4	147·2
20	35	24½	10⅝	8	3-16	40	5	1	21	4,440	222·	126·8
30	50	24½	14	8	3-16	50	7	1	21	7,750	258·3	155·
40	75	24½	14	8	3-16	50	7	1¼	21	7,750	193·75	103·33
50	100	24½	14	8	3-16	90	7	1	21	13,950	279·	139·5
60	125	24½	14	12	3-16	80	7	1¼	21	12,400	206·6	99·2
80	150	27½	17	12	3-16	80	7	1¼	24	14,080	176·	93·86
				Average		for	1in.	pipe	per	ton,	263·42	142·12
				Average		for	1¼in.	pipe	per	ton,	192·12	98·79

SUBMERGED CONDENSERS.

Ice-making Capacity in tons.	Refrigerating Capacity in tons.	Tanks. Length in feet.	Width in feet.	Depth in feet	Thickness of Iron in inches.	Number of Coils.	Pipes High.	Feet Long.	Size of Pipe in inches.	Total feet of Pipe in Condenser.	Feet of Pipe per ton, Ice-making Capacity.	Feet of Pipe per ton, Refrigerating Capacity.
5	10	10	3½	6½	3-16	9	12	7½	1	855	171·	85·5
10	20	10	7½	6½	3-16	20	12	7½	1	1,900	190·	95·
12½	25	10	7½	6½	3-16	22	12	7½	1	2,090	167·	83·6
15	30	10	8½	6½	3-16	25	12	7½	1	2,375	151·6	79·16
20	35	10	10	6½	3-16	27	12	7½	1	2,565	128·25	73·28
30	50	10	10	12½	¼	27	24	7½	1	5,130	171·	102·6
40	75	14	10	13½	¼	27	24	11½	1	7,695	191·1	102·6
60	110	14	13	13½	¼	35	24	11½	1	9,975	166·25	90·68
									Aver	age,	167·	89·

COOLING OF WATER IN PIPES EXPOSED TO AIR.

	3in. Wrought Iron Pipes.				4in. Cast Iron Pipes.			
Number of experiment ..	1	2	3	4	1	2	3	4
Temperature of the atmosphere Fahr.	58°	58°	52°·5	52°	60°	60°	60°	59°
Average difference of temperatures of the water and the air Fahr. ..	108°·7	49°·4	25°·4	14°·8	63°·8	45°·8	33°·9	27°·3
Total heat emitted per square foot per hour. Units	233·7	104·4	46·45	19·7	99·5	69·9	49·5	38·2
Heat emitted per 1° F. difference of temperature. Units	2·25	2·11	1·83	1·39	1·59	1·53	1·46	1·40

COMPARISON OF CALORIES AND BRITISH THERMAL UNITS.

The heat unit based on the centigrade or Celsius scale represents the amount of heat required to raise one kilogramme of water from 0° to 1° C. This is known as a "calorie." The British thermal unit represents the amount of heat required to raise one pound of water from 32° to 33° F.

Calories.	British Thermal Units.	Calories.	British Thermal Units.
1	3·96	60	236·16
2	7·92	70	276·12
3	11·88	75	297·
4	15·84	80	316·8
5	19·8	90	356·4
6	23·76	100	396·
7	27·72	125	495·
8	31·68	150	594·
9	35·64	175	693·
10	39·6	200	792·
15	59·4	250	990·
20	79·2	300	1188·
25	99·	400	1584·
30	118·8	500	1980·
40	158·4	750	2970·
50	198·	1000	3960·

PRESSURES IN ATMOSPHERES OF DIFFERENT REFRIGERANTS.

Temperature and Degrees. Fahr.	Sulphuric Acid. SO_2.	Methy Chloride. C_2H_3Cl.	Ammonia. NH_3	Carbonic Acid. CO_2
−22	0·37	0·76	1·13	..
—13	0·50	0·94	1·40	17·1
— 4	0·60	1·15	1·80	19·9
5	0·80	1·42	2·30	23·1
14	1·04	1·72	2·80	26·7
23	1·25	2·01	3·40	30·8
32	1·50	2·48	4·20	35·5
41	1·80	2·96	5·00	40·3
50	2·30	3·51	6·00	46·1
59	2·70	4·25	7·10	52·2
68	3·20	4·82	8·40	58·8
77	3·80	5·62	9·80	66·0
86	4·50	6·50	11·49	73·8
95	5·20	7·50	13·20	82·8
104	6·10	8·75	15·30	91·0
113	7·10	10·00	17·50	96·5

[OVER

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